N67-13874
(ACCESSION NUMBER)
155
(PAGES)
CR 80771
(NASA CR OR TMX OR AD NUMBER)
(THRU)
3
(CODE)
31
(CATEGORY)
FACILITY FORM 602

GPO PRICE $ ______

CFSTI PRICE(S) $ ______

Hard copy (HC) 5.00

Microfiche (MF) 1.00

ff 653 July 65

Accession No.

SID 66-1739

QUARTERLY REPORT NO. 2
(Aug-Oct 1966)
A STUDY OF ELECTRONIC PACKAGES ENVIRONMENTAL CONTROL SYSTEMS AND VEHICLE THERMAL SYSTEMS INTEGRATION
(Contract NAS 8-20320)

28 October 1966

D. J. Watanabe
Project Manager

Approved by:

J. A. Stevenson
J. A. Stevenson, Manager
Environmental Control
and Life Support Systems

Approved by:

W. H. T. Loh
W. H. T. Loh, Director
Aerothermo and Power Systems

NORTH AMERICAN AVIATION, INC.
SPACE and INFORMATION SYSTEMS DIVISION

Accession No.

SID 66-1739

QUARTERLY REPORT NO. 2
(Aug-Oct 1966)
A STUDY OF ELECTRONIC PACKAGES ENVIRONMENTAL CONTROL SYSTEMS AND VEHICLE THERMAL SYSTEMS INTEGRATION
(Contract NAS 8-20320)

28 October 1966

D. Watanabe
D. J. Watanabe
Project Manager

Approved by:

J A Stevenson
J. A. Stevenson, Manager
Environmental Control
and Life Support Systems

Approved by:

W. H. T. Loh
W. H. T. Loh, Director
Aerothermo and Power Systems

NORTH AMERICAN AVIATION, INC.
SPACE and INFORMATION SYSTEMS DIVISION

FOREWORD

This document is submitted by the Space and Information Systems Division (S&ID) of North American Aviation, Inc. to the George C. Marshall Space Flight Center (MSFC) of the National Aeronautics and Space Administration in accordance with Contract NAS 8-20320. This report summarizes the study activity conducted during the second quarter of the contract.

TECHNICAL REPORT INDEX/ABSTRACT

ACCESSION NUMBER	DOCUMENT SECURITY CLASSIFICATION
	Unclassified

TITLE OF DOCUMENT	LIBRARY USE ONLY
QUARTERLY REPORT NO. 2, A STUDY OF ELECTRONIC PACKAGES ENVIRONMENTAL CONTROL SYSTEMS AND VEHICLE THERMAL SYSTEMS INTEGRATION	

AUTHOR(S)
Watanabe, D. J.

CODE	ORIGINATING AGENCY AND OTHER SOURCES	DOCUMENT NUMBER
NAJ65231	Space and Information Systems Div. of NAA, Downey, California	SID 66-1739

PUBLICATION DATE	CONTRACT NUMBER
28 October 1966	NAS 8-20320

DESCRIPTIVE TERMS

Environmental Control
Thermal Control
Thermal Analysis

ABSTRACT

Study to determine the optimum environmental control (passive or active) systems for thermally conditioning individual electronic packages for space missions of durations ranging from $4\frac{1}{2}$ hours to 180 days. The environmental control system shall be optimized on the bases of mission duration, operational temperature limits and heat dissipation rate of the electronic package.

This quarterly progress report describes the additional data to augment the basic requirements and constraints, the survey effort in determining the developmental trends, the parametric study of thermal control processes and components, and the synthesis and reliability analysis of integrated systems.

FORM M 131-V REV 1-66

CONTENTS

ILLUSTRATIONS

TABLES

1.0 INTRODUCTION AND SUMMARY

This study is directed toward defining environmental control system (ECS) concepts to meet the increased complexity and changing thermal conditioning requirements of the electronic equipment on the Saturn V vehicle for missions of durations varying from four and one-half hours to one-hundred eighty days. The study objective is to establish the optimum environmental control concepts for thermally conditioning individual electronic packages. Essential characteristics of the optimum environmental control systems are maximum practicable reliability, maximum operating range, minimum weight, minimum power, and minimum volume. The optimum systems are to be selected on the basis of future Saturn mission requirements, future trend in the design and development of the electronic equipment and being an integral part of the overall vehicle thermal system.

The results of the study effort will help to define the developments and advancements required in the technology of thermal control methods and systems. Critical areas or pacing items for development are to be identified. This should help to insure the necessary development to be accomplished and on a timely basis. Other results from the study include design guidelines for electronic packages to achieve optimum environmental control designs and for integrated ECS and vehicle thermal control systems.

Longer mission duration and greater heat dissipation requirements together with closer temperature regulation require thermal systems in which the control method is self regulating and is capable of operating under widely varying conditions. The simple passive system cannot meet these requirements. The study effort is directed toward the investigation of system concepts which can function reliably within widely varying conditions. These systems concepts, like the simple passive system, will depend upon rejecting waste heat to space, but the means of transporting heat from the source to the radiating surface and the surface itself will be quite different. The systems will be more complex than the pure passive approach and thus it becomes a challenge to achieve practical, simple and reliable systems. Many alternate concepts are to be investigated and the most promising ones are to be studied in depth.

SUMMARY

During this reporting period, satisfactory progress has been made in the study effort in meeting schedules and in performing the planned tasks. Figure 1 indicates the schedule position for the major tasks performed. The work in Tasks 1, 2 and 3 have been continued from the previous quarter and Task 4 has been initiated during this reporting period.

In Task 1, additional data have been provided to augment the basic requirements and constraints established during the previous quarter. These include revised estimate for power penalty, future Saturn vehicle configurations, and some design details of the instrument unit structure. Additional information on astrionic equipment is presented, which is based on survey of future development in astrionic equipment and thermal control components. Several tentative conclusions have been reached with regards to astrionic equipment. The overall heat load will remain relatively constant because the large power consumming equipment are least miniaturizable. Also, heat load tends to remain constant due to increased equipment functions and complexity. Another conclusion is the indicated trend toward integrated astrionic equipment and coldplates. This arrangement results in weight savings, improved coolant utilization and improved operating temperature with resulting improvement in reliability. There are indications that the operating temperatures and operating range may be increased without reduction in reliability by improved manufacturing techniques. The cumulative effect of these trends means that the environmental control requirements remain basically the same and possibly easing to some extent.

Since future projections of heat load profiles are somewhat nebulous, an analysis was made of existing heat load profiles which indicated that it is possible to use hypothetical heat load profiles plotted on probability paper to represent the complete range of possible load profiles. In addition, the heat load profile plotted on probability paper provides a convenient basis to determine the best combination of temperature control methods to meet peak or spike loads and the substained heat loads.

A review of astrionic equipment package thermal design and expendable cooling methods has been made. This included internal heat transfer and temperature control devices and techniques for electronic packages and various expendable cooling methods, coolant sources, design and control concepts.

As a continuing effort, the parametric study was conducted in several areas. These included: the application of thermoelectric cooling and temperature control, a detailed analysis of the utilization of hydrogen boil-off, space radiator design and analysis, temperature regulation methods, and heat sinks. Based on the analysis conducted and the assumptions made, it is concluded that the utilization of hydrogen boil-off is feasible and

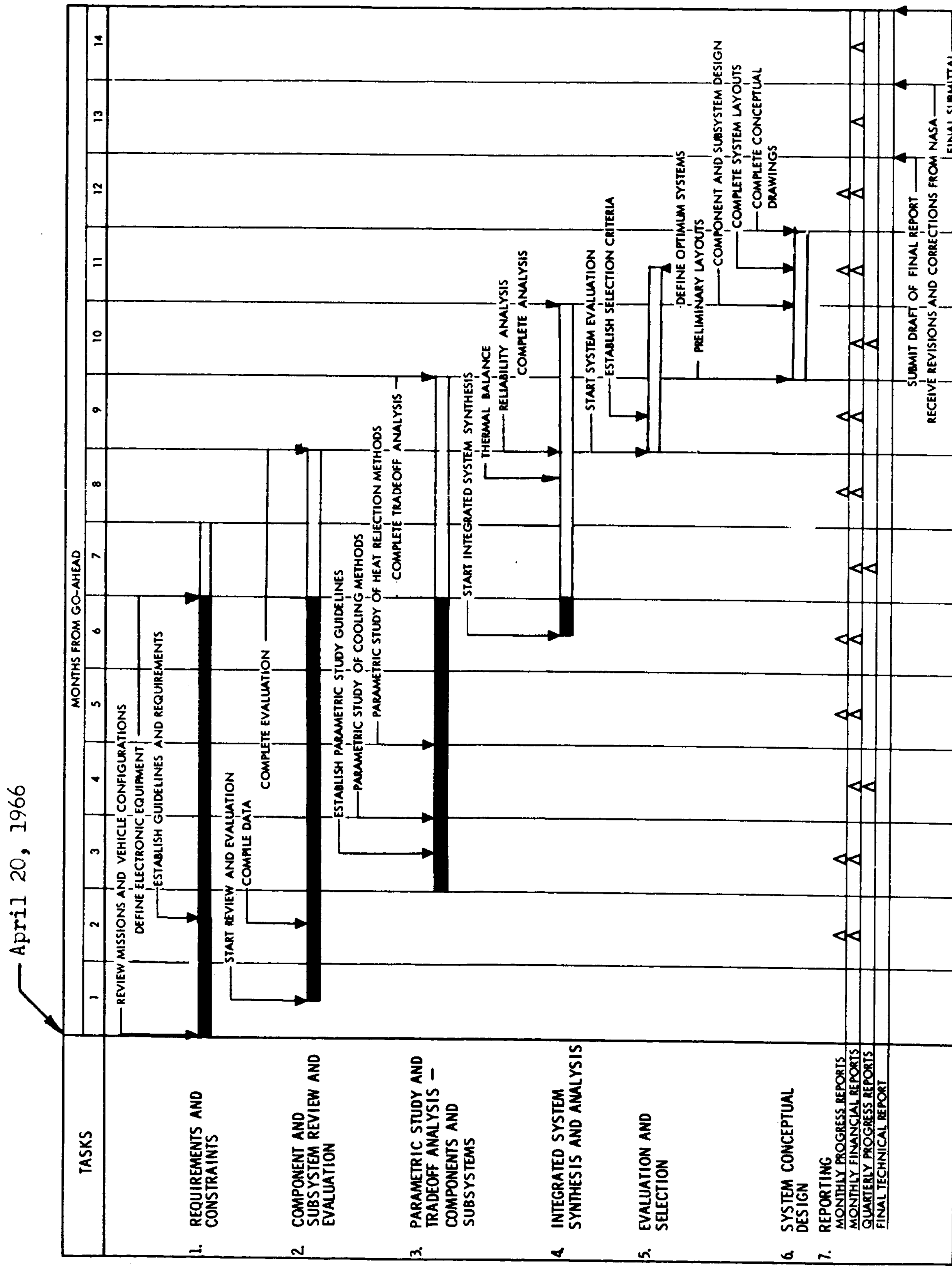


Figure 1. Program Schedule

appreciable weight savings are possible by a combination of hydrogen-gas-fluid cooler and space radiator. The study of heat sinks has been and will continue to be directed toward establishing the regions of applicability in terms of heat load capacity and duration. This effort is considered to be quite significant because it provides a readily useable means for selecting the necessary heat sink for a particular application.

The task on integrated system synthesis and analysis was initiated during this period and will be continued. A number of system concepts are described with regard to some general characteristics. These and additional concepts will be investigated in detail during the next period. Some effort has been devoted to system reliability because of the importance of reliability in the synthesis and analysis of the integrated thermal systems. This effort will be expanded during the next period.

This quarterly report is intended to serve not only as a progress report but to be a reference document. For this reason, it contains data reproduced from various references and detailed analysis in Appendix A.

Based on the results and progress to date, no change in the study plan appear necessary in order to satisfactorily meet the objectives of the study contract.

2.0 CURRENT STUDY STATUS

2.1 SYSTEM REQUIREMENTS AND CONSTRAINTS

Detailed data have been included in certain areas to better define the requirements and constraints that may be imposed on the temperature control system and astrionic equipment for future instrument instrument unit or units. The data provide the range of design requirements or conditions that should be considered in this study.

2.1.1 Incident Heat Loads

For the future space missions indicated in the first quarterly report, Reference 1, the various vehicle orientations and orbit paths have been established and are illustrated in Figure 2. These vehicle and orbit paths have been used to establish the incident heat loads on the various surfaces of a simulated space vehicle. This provides the range of maximum and minimum incident heat loads to be used in this study.

The computation was accomplished with the aid of a computer program described in Reference 2. The equations used in the computation is summarized in Table 1. The equation for the incident heat from planetary emission is based on the assumption that the surface emission is constant over the entire planet surface. This appears to be reasonably accurate for the case of the earth surface. For the lunar surface, the variation of the surface radiation is incorporated into the computer program to provide reasonably accurate incident heat loads from this source.

Incident heat loads for the following cases have been computed:

a) Planet oriented: circular-near earth
circular-near lunar

b) Solar oriented: circular-near earth
circular-near lunar

c) Tangent-to-flight path: circular-near earth
synchronous earth orbit
circular-near lunar

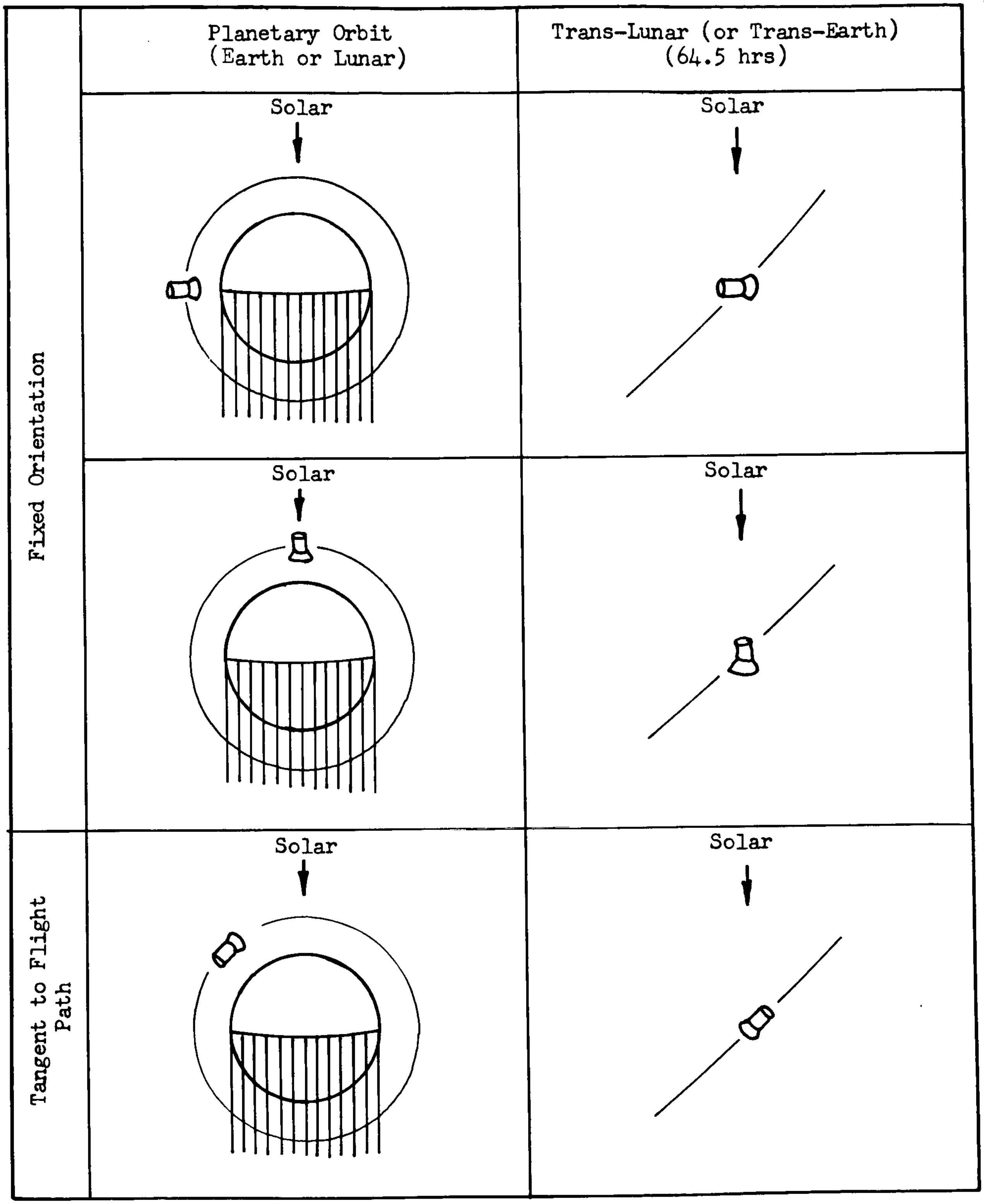


Figure 2. Vehicle Orbit and Orientation

Table 1. Equations for Incident Heat Flux and Incident Heat

Heat Sources	Incident Heat Flux Btu/(hr)(sq ft) (Ref. 2)	Incident Heat, Btu/hr (Ref. 2)	Incident Heat, Btu/hr Flat Plate
Direct Solar	$S F_S$ $F_S = \cos \theta_N$	$Q_S = \alpha_S A S F_S$	$Q_S = \alpha_S A S \cos \theta_N$
Reflected Solar	$S r_E F_R$ $F_R = F_E \cos \theta_S$	$Q_{SR} = \alpha_S S r_E F_R A$	$Q_{SR} = \alpha_S A S r_E \left[1 - \sqrt{1 - \left(\frac{r}{r+h}\right)^2}\right] \cos \psi$
Planetary Emission	$S\left(\frac{1-r_E}{4}\right) F_E$	$Q_{EE} = \epsilon A S \left(\frac{1-r_E}{4}\right) F_E$	$Q_{EE} = \epsilon A (1-r_E) \sigma \left(T_P^4 - T_S'^4\right)$ $\times \left[1 - \sqrt{1 - \left(\frac{r}{r+h}\right)^2}\right] \cos \beta$

2.1.2 Spacecraft Power

In the first quarterly report, the assumed power penalty range was given as 200 to 500 pounds per kilowatt. Recent data presented in Reference 3, indicates that this range to be lower than what may be expected for future spacecraft power systems. Figure 3 from Reference 3 gives the system weight and power output for a number of power systems. On the basis of this data, a power penalty of 240 to 1500 pounds per kilowatt may be more realistic and will be the values used in the study.

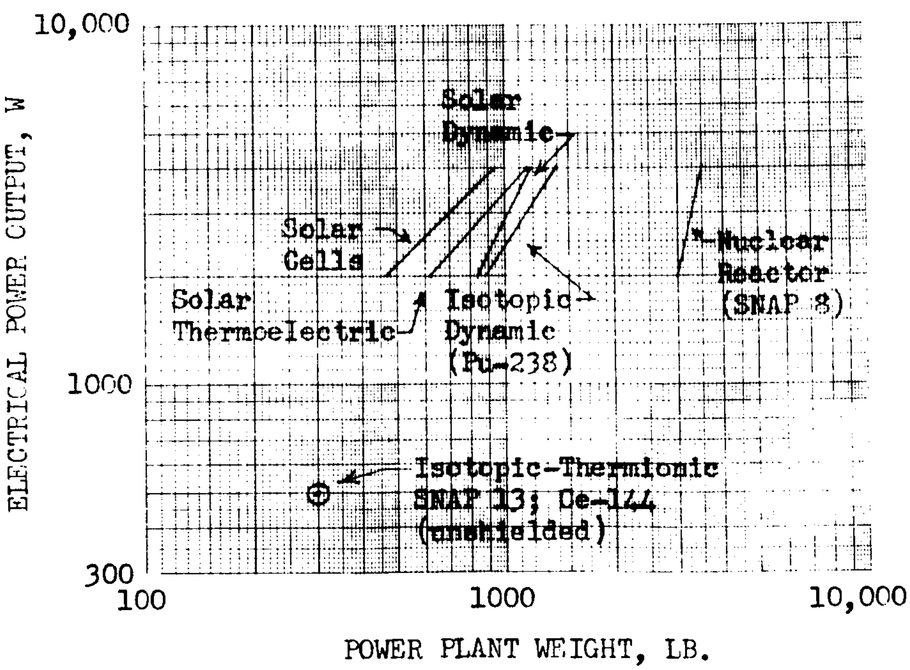


Figure 3. Future Spacecraft Power Sources

2.1.3 Vehicle Configuration

It has been assumed that the future vehicles for the missions and mission durations selected for the study will be the Saturn V with various combinations of up-rated upper stages. Figures 4 and 5 from Reference 4 illustrates a possible configuration in which several instrument units are utilized. It may be assumed that all the instrument units are not identical and that the type, the number, and the duty cycle of the astrionic equipment may vary from a bare minimum to one which would perform multi-functions on a continuous or intermittent basis. This suggests that no single thermal system design would be adequate for all the instrument units used in this vehicle configuration.

Another possible vehicle configuration is illustrated in Figure 6 from Reference 5 in which the instrument unit is attached to a modified Saturn-II stage. In this application, the possibilities of utilizing the residuals are indicated. The thermal control system design could incorporate means for utilizing the residuals directly or the by-product of a process which utilizes the residuals such as fuel cells which produce water as a by-product.

Other possible configurations and applications of the instrument unit will be considered as information becomes available from future mission and vehicle studies.

2.1.4 Instrument Unit Design

Although the structural details of the instrument unit are not pertinent to this study, some indication of the possible future design is considered useful for the conceptual design of the thermal control system. Figures 7 and 8 illustrate possible design concepts and Figure 9 gives a list of leading lightweight structural materials. These data are useful in the consideration of the possible integral design of the space radiator and the vehicle or instrument unit structure. The surface area and volume limitations and other possible physical constraints are of significance to this study. Conversely, the effect of the thermal control system design on the structural design is of interest and will be examined during the study effort.

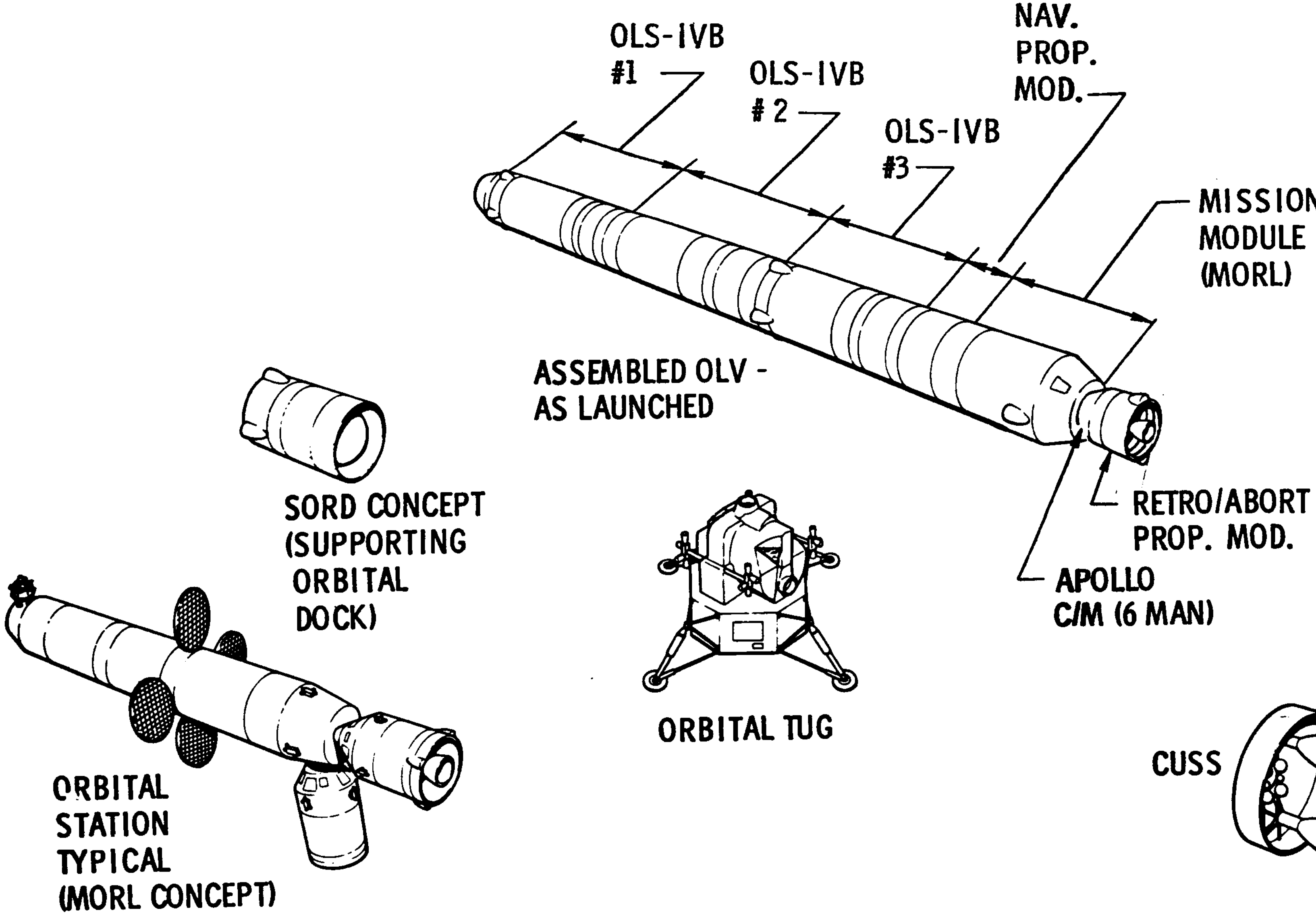


Figure 4. Orbital Launch Vehicle Configuration

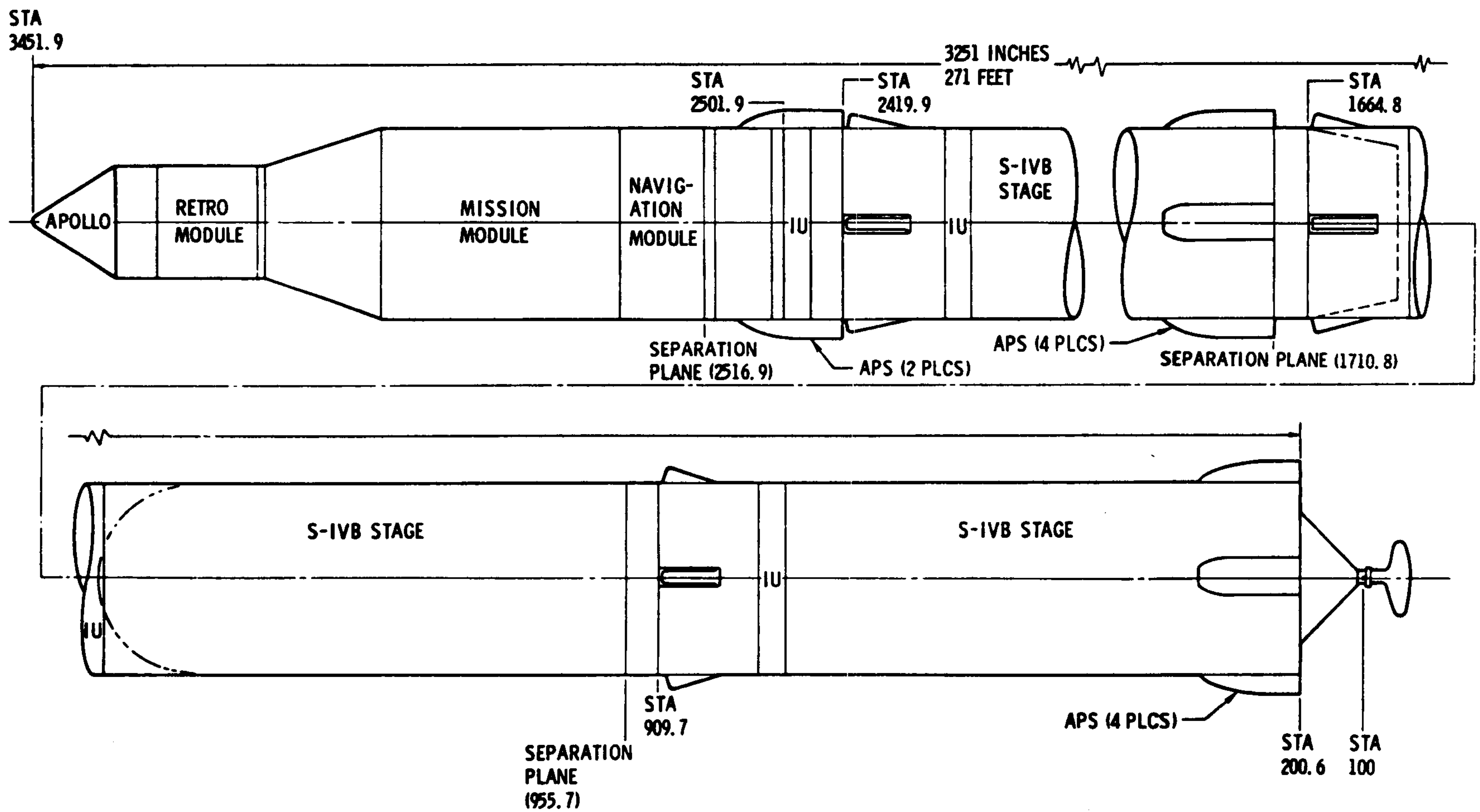


Figure 5. Orbital Launch Vehicle Configuration

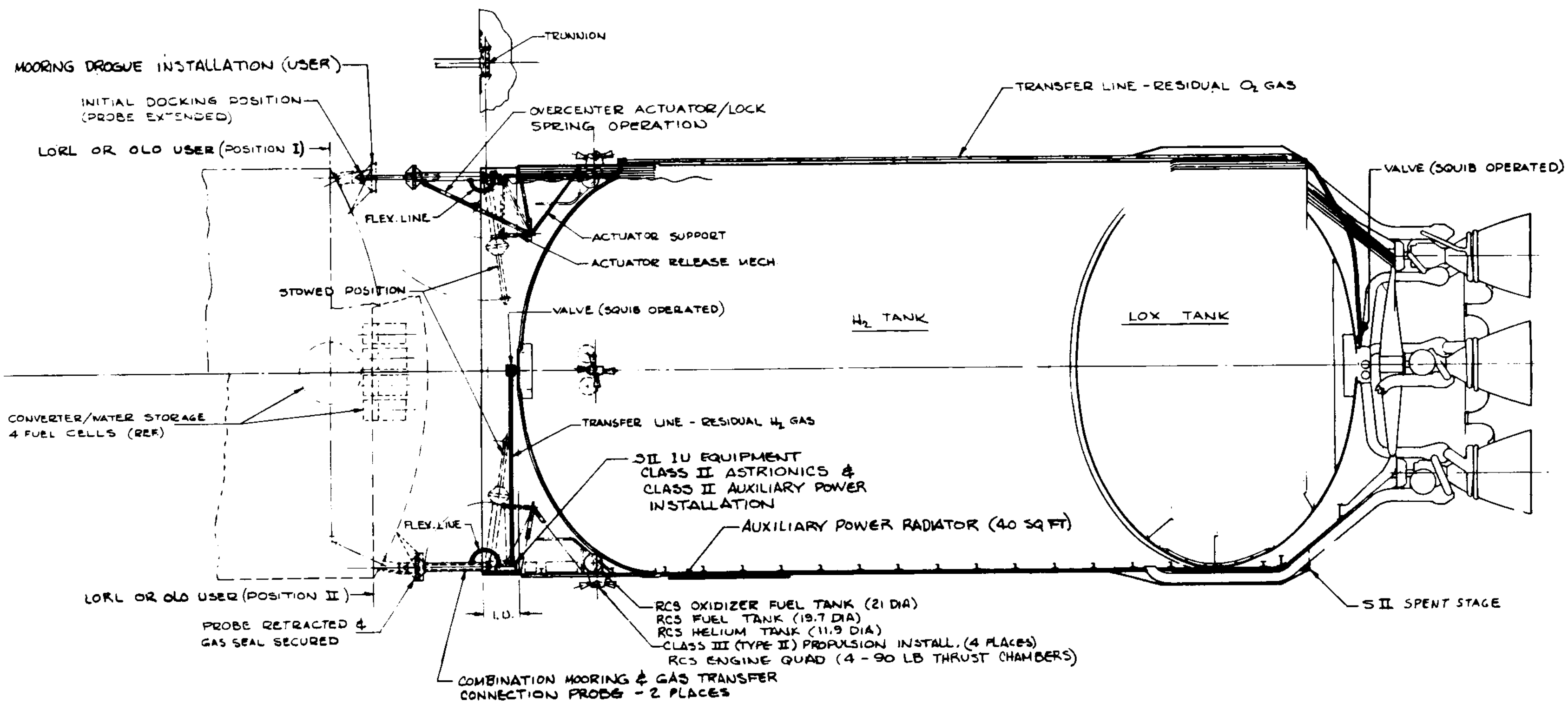


Figure 6. Uprated S-II Configuration

NONDIMENSIONAL DESIGN PARAMETERS

$$\frac{R}{t} = \frac{130}{0.116} = 1120$$

$$\frac{L}{R} = \frac{33.5}{130} = 0.258$$

$$Z_L = \left(\frac{L}{R}\right)^2 \left(\frac{R}{t}\right) \sqrt{1 - \mu^2} = 74.5$$

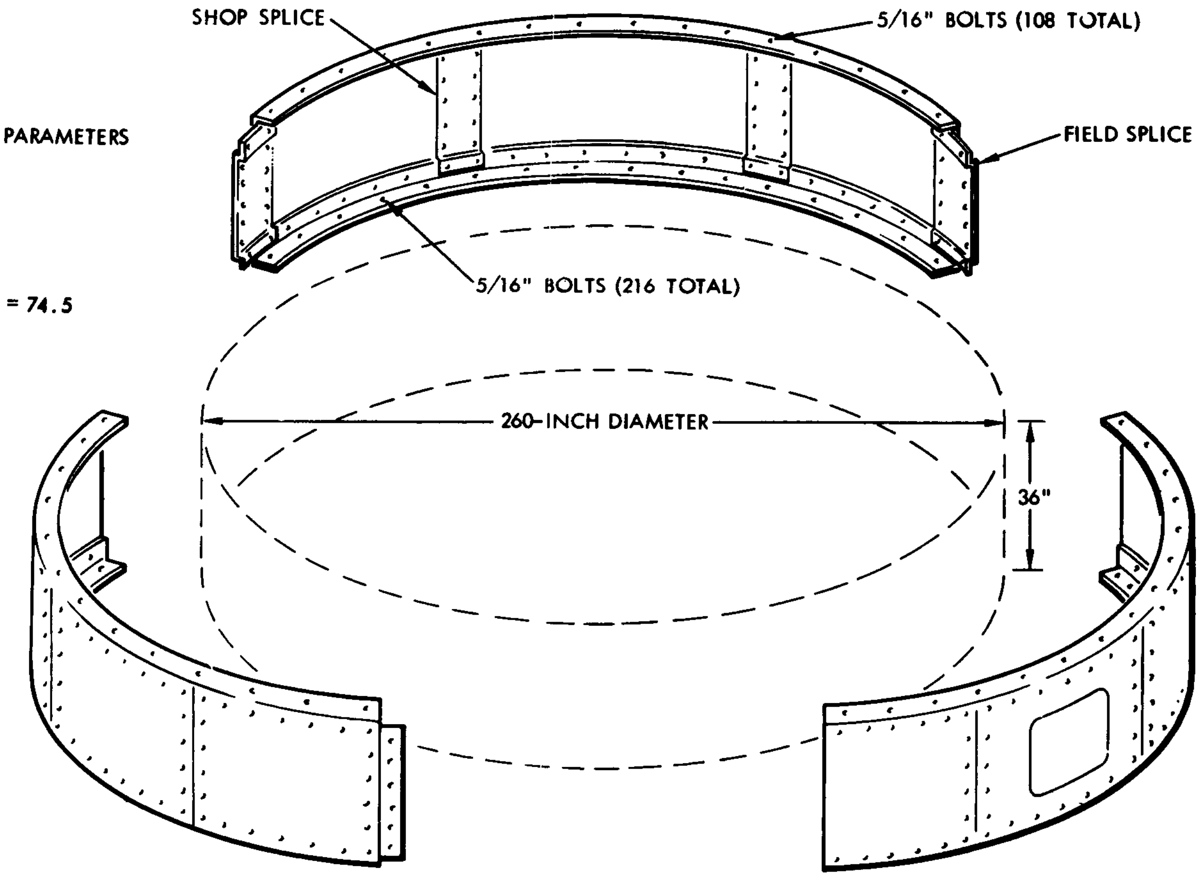


Figure 7. Typical Shell Structure For Instrument Unit

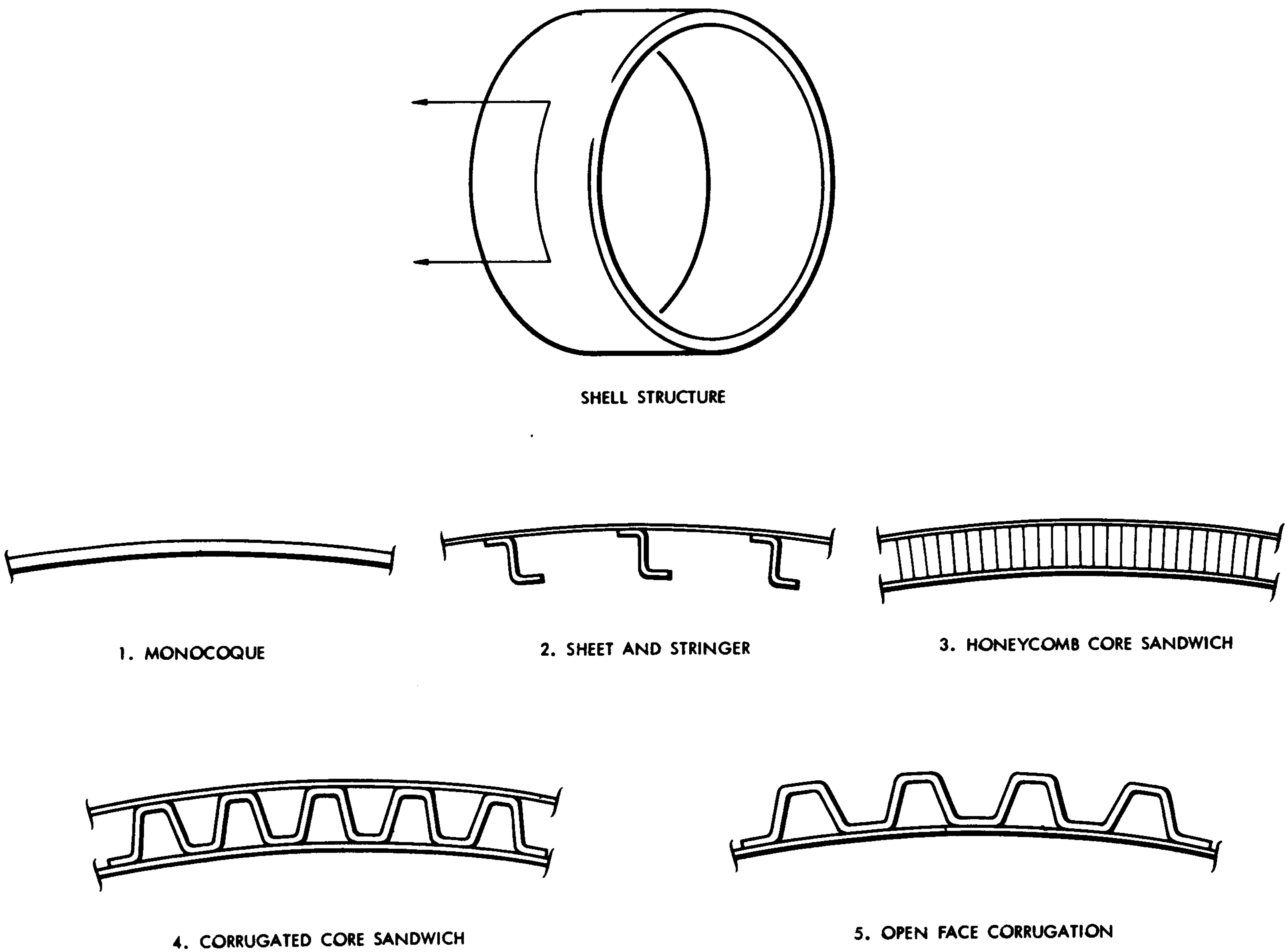


Figure 8. Typical Types of Construction for Lightweight Shells

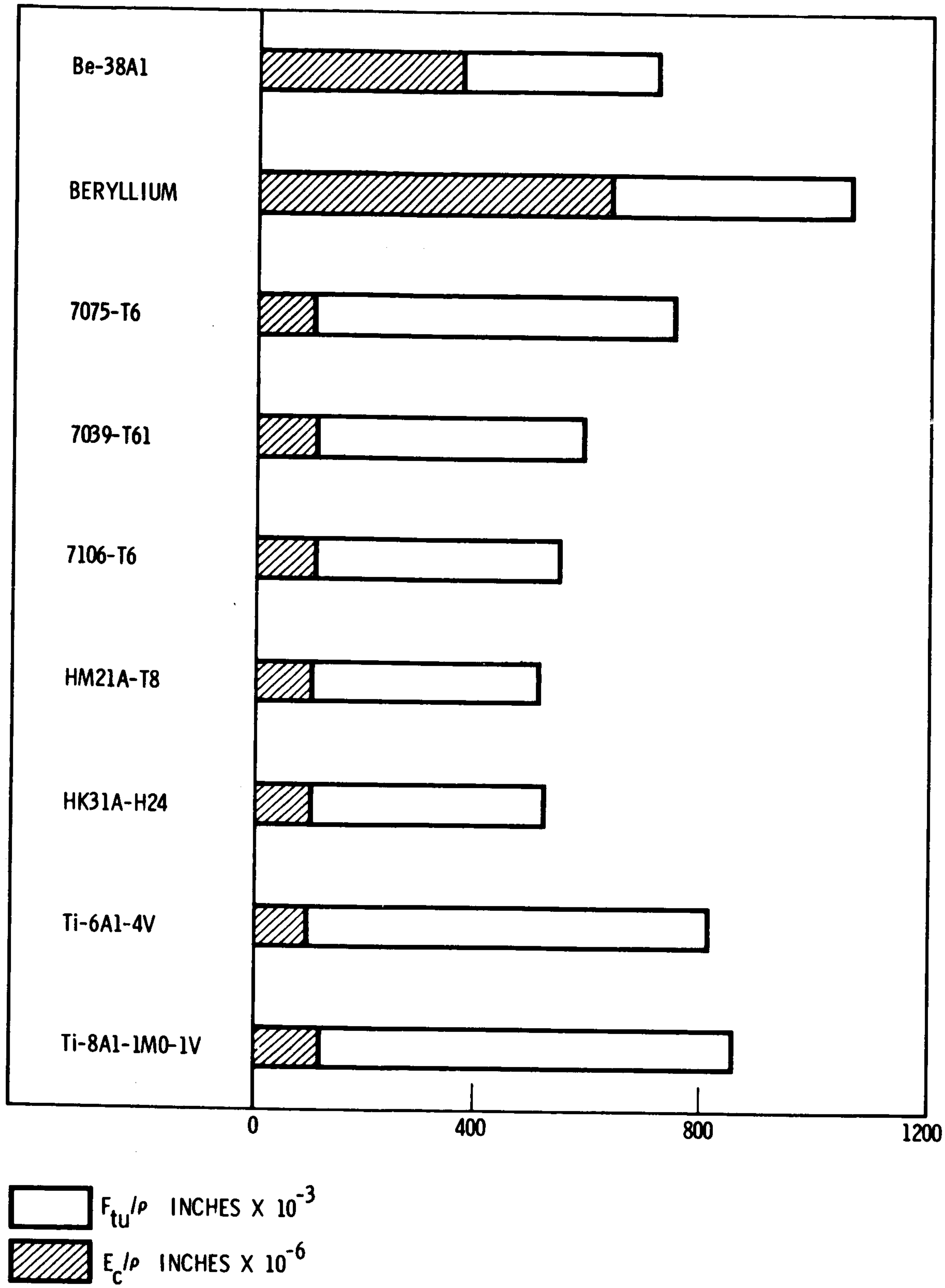


Figure 9. Strength-to-Density and Elastic Modulus-to-Density Ratios for Leading Lightweight Structural Alloys

2.1.5 Heat Load Analysis

Heat dissipation loads or requirements of astrionic equipment can be expected to vary over a relatively wide range, particularly for the longer duration missions. The variations in the form of short duration spikes or peaks and possible wide range between the maximum and minimum heat dissipation values require a careful analysis of the heat load profile. The complex nature of the heat load profile makes it difficult to properly size the cooling system and to determine whether or not it would be beneficial to use auxiliary cooling methods to augment the primary cooling system. To simply use a mean average heat load is not adequate because it precludes the consideration of the various possible combinations of cooling techniques to achieve minimum weight, power and volume. For example, the combination of an expendable cooling method and a recycle-closed loop system may be far superior to a somewhat less complex system which consists of only a recycle-closed loop, but overdesigned to meet the transient or peak loads. The over-design or the increased capacity from the steady load design could result in excess capacity for major portion of the operating time and thus impose added penalties due to increased weight and possibly inefficient operation.

To determine a rational approach to analyzing the heat load profile, a heat load analysis was made and the results are given and briefly discussed.

To conduct the analysis, the AES equipment load profiles (Reference 6, pp. 135-136) and Apollo primary loop load profiles (Reference 7, pp. 32-38) were used. In general, the electronic load profiles will be nearly identical to the required heat dissipation profile since nearly all of the energy appears as heat.

Data from Reference 7 have been plotted in Figure 10. Figure 10 shows a number of short duration spikes of considerable magnitude due to intermittent use of equipment and longer duration periods of elevated power (heat dissipation) of lower magnitudes. All are superimposed on a steady load. These basic characteristics are likely to be present in any electronic equipment systems for space operation. Figure 11 shows the

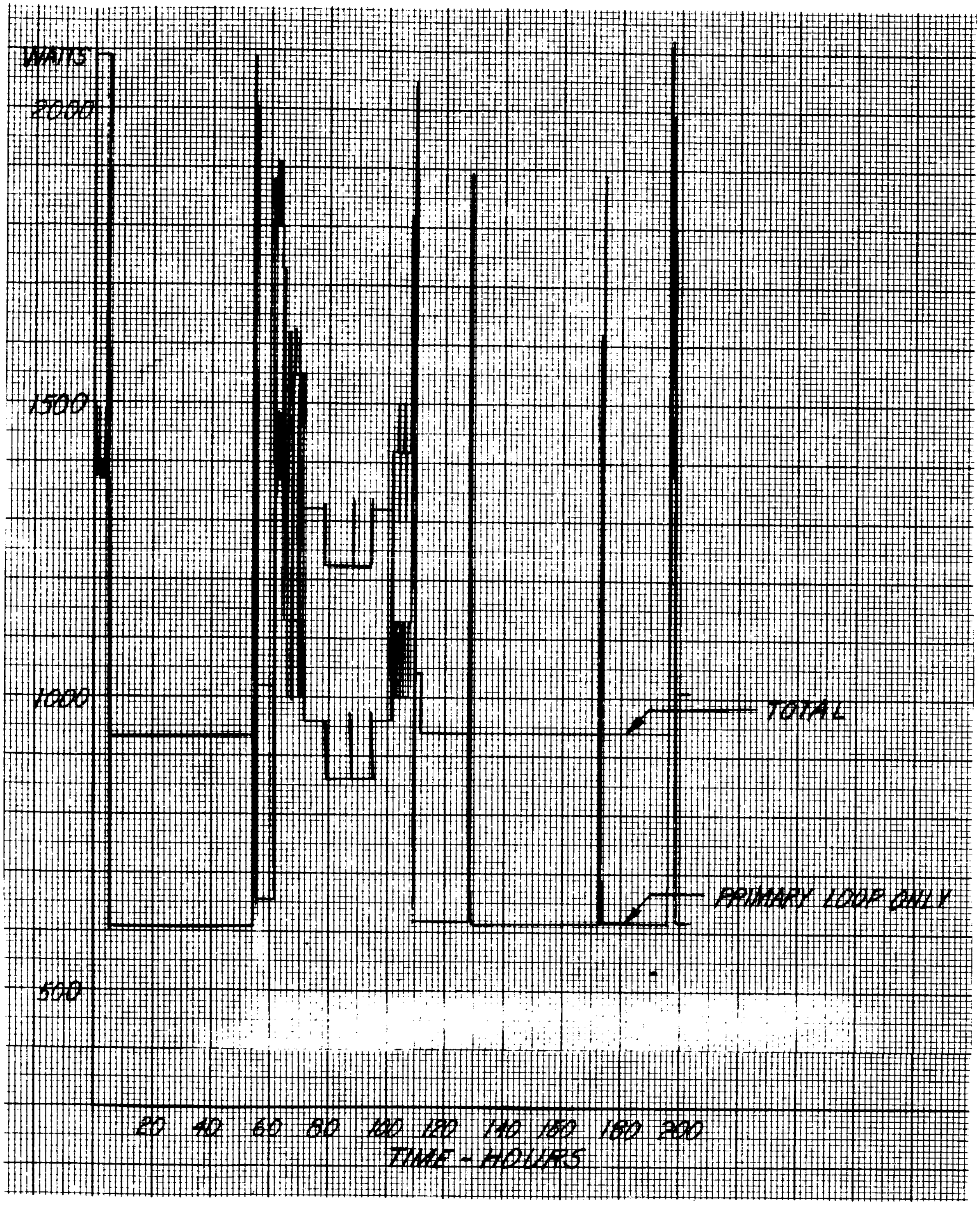


Figure 10. Load Profile, Block II Apollo Coldplate Network

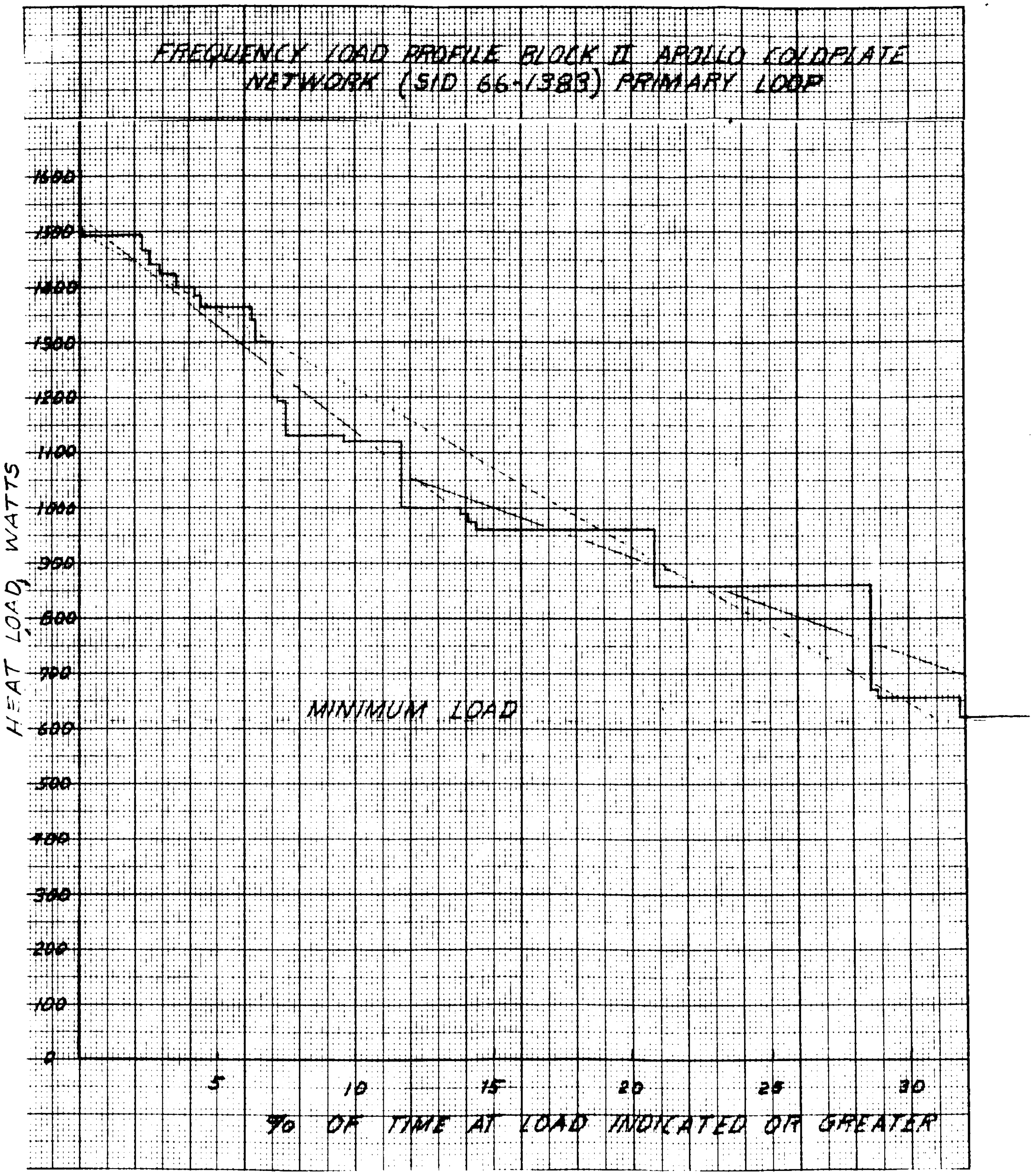


Figure 11. Frequency Load Profile, Block II Apollo Coldplate Network

frequency load profile of Figure 10. This frequency load profile is also plotted on probability paper, Figure 12. Figure 12 indicates the duration of the maximum peak loads to be about 2% of the total time and a steady load of about 600 watts for a period of about 70% of the total time. The intermediate loads between the maximum and the steady loads are indicated by a series of steps and it appears that a straight line could be a fair representation of these loads. A load plot such as this provides a convenient means for determining the load levels and the approximate durations which can be used to make a weight trade-off for various combinations of cooling methods. The application of this data is discussed below and in the section on the utilization of hydrogen vent gas.

The heat load data given in Reference 6 and 7 have been summarized in Table 2 along with the average load to maximum load ratio and the minimum load to average load ratio. The variations in these ratios for the various mission periods provided a basis for estimating the approximate values for these ratio for longer durations. For the longer missions, the average load to maximum load ratio of 0.25 and the minimum load to average load of 0.75 are assumed and used below.

The above data was used to provide the basis for constructing a hypothetical load profile to be used in the analysis of the utilization of the hydrogen vent gases. A maximum load of 2,000 watts was assumed and based on this, an estimated average load of 500 watts was established. However, 500 watts may be too optimistic for mission durations less than 50 days, so 550 watts may be a better estimate for 0 to 50 day duration missions. Using these values and the assumed ratios given in the above paragraph, a hypothetical load profile is plotted in Figure 13.

The analysis and data presented in the above paragraphs provide a basis for establishing load profiles in the absence of detailed heat load data. It would be possible to establish a set of load profiles, such as Figure 12 or 13, which represent possible heat loads to be expected for future missions. This can be done by assuming a range of values for the maximum, minimum and average heat loads and various percentages of time at the maximum and minimum heat loads. This would establish the slope of the line connecting the maximum and minimum heat loads and this line would represent the intermediate heat loads as illustrated in Figure 13.

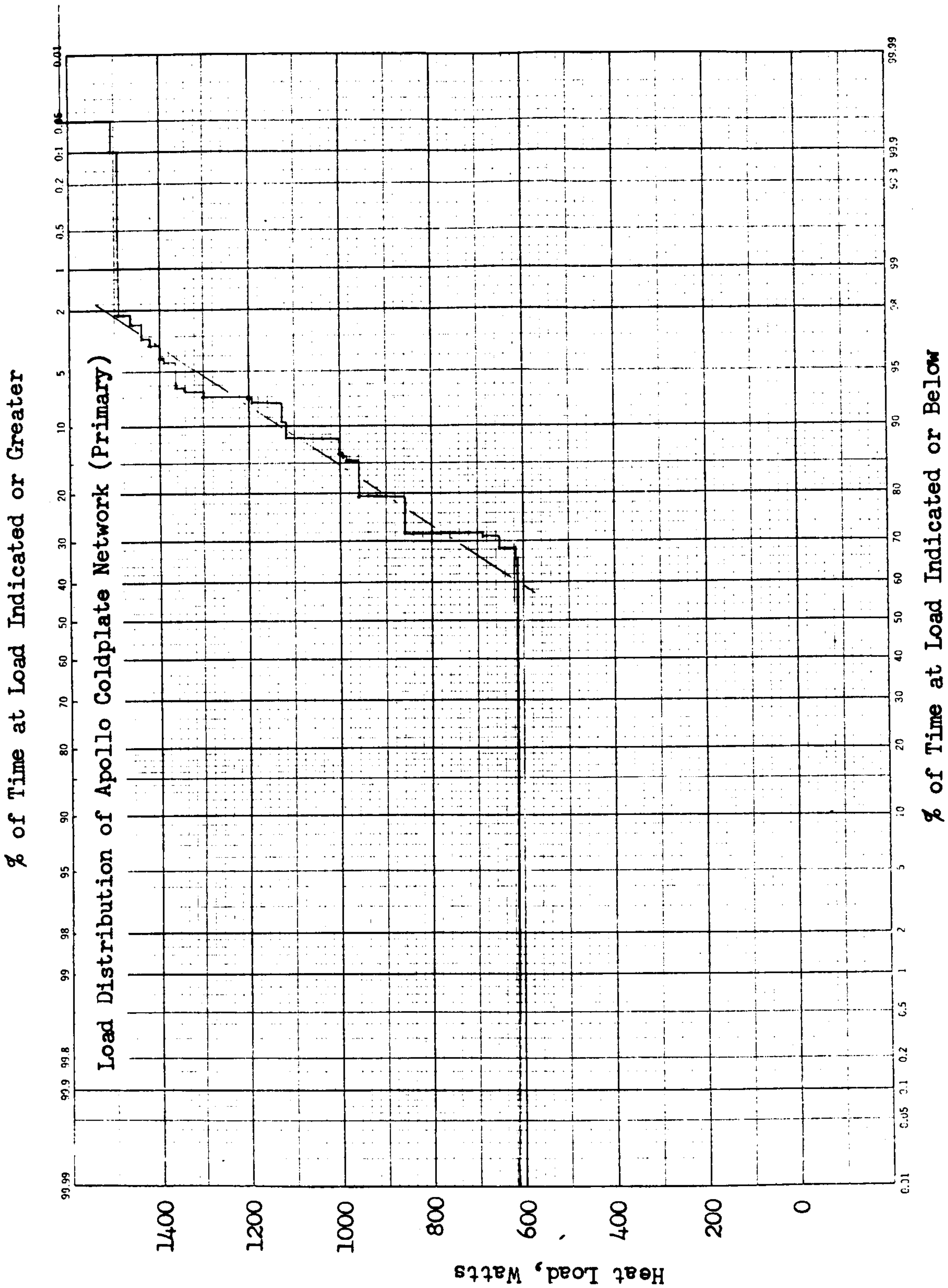


Figure 12. Load Distribution of Block II Apollo Coldplate Network

Table 2. Heat Load Summary

Mission Period	Heat Load (watts)			Ave. Load / Max. Load	Min. Load / Ave. Load
	Maximum	Minimum	Average		
1. Earth Polar Orbit	580.8	431.8	444.7	0.77	0.96
2. Earth Synchronous Orbit	910.0	431.8	463.3	0.51	0.93
3. Lunar Polar Orbit	1560.3	455.8	481.2	0.31	0.94
4. LEM Escort	1389.3	460.0	416.1	0.37	0.89
5. Trans-Lunar Coast	1483.5	412.2	506.7	0.34	0.81
6. Trans-Earth Coast	1483.5	412.7	506.7	0.34	0.81
7. Apollo:					
Primary equipment loop	1620	612	745		
CM, C & N timeline (PSA unit)	175	16	36.9		
Insertion timeline	370	309	319		
Total	2165	937	1101	0.55	0.85

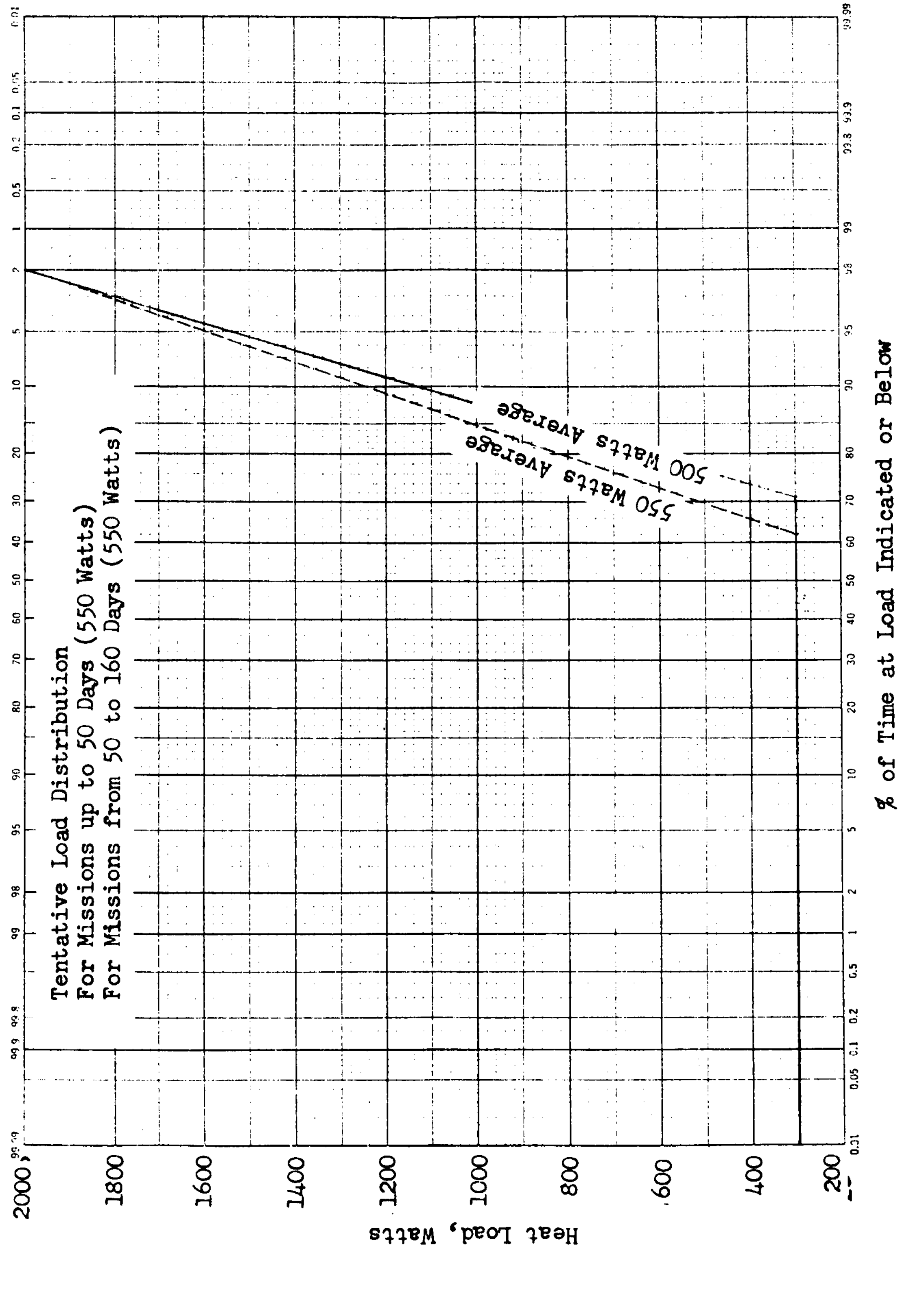


Figure 13. Tentative Load Distribution

During the coming quarter, a set of load profiles will be established to be used in determining the necessary systems which may range from the simple open or expendable system to the more complex system composed of both expendable and closed-recycle system.

2.1.6 Astrionic Equipment

Equipment Functions

The equipment functions, i.e., the generic equipments, required for the postulated missions may be grouped into the following categories: (1) Guidance and Navigation, (2) Communications, and (3) Data Management. For the purpose of this study, the primary power supply is not considered astrionic equipment and is treated separately. Also, primary mission experiments and sensors are not included since these are special equipment whose requirements could vary widely. Table 3 lists the equipment functions anticipated for a large earth or lunar orbiting vehicle.

The equipment functions listed in Table 3 should not be considered final. A final list, in practice, is selected for each type of mission which is peculiar to that mission from exhaustive trade-off studies and cannot be performed as part of this study. These studies utilize such data as individual equipment's reliability, size, weight and power, the required probability of mission success, and the availability of other modes of operation. Therefore, the list should be considered only typical. Equipment/equipment functions which require redundancy or backup are indicated by an asterisk. Again, the final redundancy and back-up requirements are determined at the same time as the basic equipment selection for each specific type of mission by extensive trade-off studies. Whether the vehicle is manned (or capable of being manned) in orbit also influences significantly the equipment functions required; e.g., the need for displays and the need for redundant or backup equipment if manual backup is available.

Typical Astrionic Equipment Characteristics

Table 3 gives typical heat load values of astrionic equipment packages. Their values are also typical since, in many cases, one equipment can influence another. For example, the size and capability of a navigation computer is influenced by the choice of IMU type., i.e., strapdown, inertial (gimballed) or stellar-inertial. Furthermore, heat loads of individual electronic packages are also dependent on other performance parameters such as reliability and equipment precision. Also, a

Table 3. Typical Astrionic Equipment and Heat Load Values

Equipment / Characteristics	Heat Load Polar Earth Orbit (Watts)		Heat Load Syn Earth Orbit (Watts)		Heat Load Lunar Orbit (Watts)	
Guidance and Control	Present	Future	Present	Future	Pres	Future
*IMU	65	35	65	35	170	80
*Digital Computer	100	10	100	10	200	20
Star Tracker	15	10	15	10	15	10
*Horizon Sensor	20	15	20	15	20	15
Rendezvous Radar	50	40	60	40	300	200
Control System Transformer	10	5	10	5	10	5
*Autopilot Sensor	65	50	65	50	65	50
*Autopilot Computer	50	10	50	10	50	10
Radar Altimeter	70	50	70	50	70	50
**Displays	20	20	20	20	20	20
**Controls	10	10	10	10	10	10
Radio Transponder	30	20	30	20	100	60
Communications						
*Telemetry	10	5	10	5	50	20
*Transponder	40	20	40	20	80	40
*Transceiver, Voice/Data	100	50	100	50	300	200
*Antenna	0	0	0	0	0	0
*Tracking Beacon	20	10	20	10	50	30
Data Management						
*Signal Conditioner	20	2	20	2	20	2
*Multiplexer	4	4	4	4	4	4
Switching and Routines	5	5	5	5	5	5
Computer	50	5	50	5	100	10
Recorder	20	15	20	15	40	20
Buffer Store	20	5	20	5	40	10

* Requires Redundancy or Backup
** Manned or Manable Vehicles Only
Future - Available in the Early 1970's

package heat load depends to a certain extent on additions to the basic functions as the use of higher rated devices, self-test capabilities, and redundant parts. Mission total heat loads are not shown on Table 3 since these would be meaningless without specifying the final equipment list, including equipment redundancy and backup. Table 4 shows typical thermal characteristics (temperature and temperature regulation requirements) for various astrionic equipment packages. Again, as in the package heat load, the values specified are typical. For a given equipment package, wider tolerable temperature range and temperature regulation requirements can be traded off with increased equipment complexity by the use of higher rated parts, lower temperature coefficient devices, and temperature compensation circuitry.

Typical Astrionic Equipment Component Characteristics

Table 5 shows the thermal characteristics of various active components which make up astrionic packages. Again, the values of the component characteristics are typical and, in actual design, are determined to a large extent by the manner of their usage in the particular application.

High temperatures result in performance degradation and accelerate failure rates for parts and components. As noted in Table 5, the maximum operating temperature of most components used in astrionic equipment is dictated by reliability considerations. This is due to the large number of active electronics and devices used, each of which invariably exhibits an increasing failure rate with higher temperatures. Figure 14 shows the failure rate acceleration factor with operating temperature (Junction) for an integrated circuit. Many such circuits are used in present day computers and space electronics. Failure rate versus operating temperature of other typical components are given by Figures 15 through 18. Additional failure rate data for thin film chips, ceramic printed circuits, and integrated circuits are given by Tables 6 and 7. These illustrate that reduction in upper temperatures (down to about 20°C) is reflected in significantly lower failure rate. Since these failure rates are additive in computing the circuit and the package failure rate, it should be apparent that the reliability is almost entirely dependent on the existing thermal environment, the thermal environmental controls provided, and the package internal thermal design.

Table 4. Astrionic Equipment Typical Thermal Characteristics

Astrionic Package Type	Tolerable Ambient Temp Range (°F)	Tolerable Operating Interface Temperature		Primary Internal Heat Transfer Method	Internal Temp Control Req'd (°F)
		Range (°F)	Location		
Digital Computers	-65 to 160	-65 to 120 55 to 125	(1) Coolant (2) Vehicle Coldplate	Conduction Conduction	None None
IMU	-65 to 180	-65 to 120	(1) Coolant	Conduction or Forced Conv.	±1
Star Trackers	-65 to 160	-65 to 160	Mounting	Conduction, Rad.	±20
Radars	-65 to 160	-65 to 160	(1) Coolant	Conduction	None
Radar Altimeters	-65 to 160	-65 to 160	(1) Coolant	Conduction	None
Displays	-65 to 160	60 to 120	Mounting	Cond, Rad	None
Cockpit Controls	-65 to 160	60 to 120	Mounting	Conduction	None
Transponders	-65 to 200	-65 to 120	Mounting	Conduction	None
Telemetry	-65 to 160	-65 to 120	Mounting	Conduction	None
Transceivers	-65 to 160	-65 to 120	(1) Coolant	Conduction	None
Signal Conditioners	-65 to 160	-65 to 120	Mounting	Conduction	None
Recorders	-65 to 180	-65 to 120	Mounting	Conduction	None
Autopilot Sensor	-65 to 160	-65 to 120 -65 to 125	(1) Coolant (2) Vehicle Coldplate	Conduction Conduction	None
Horizon Scanners	-65 to 160	-65 to 160	Mounting	Conduction	None
Antennas	-110 to 250	-110 to 200	Mounting	Conduction	None
Battery	-65 to 140	50 to 110	Coolant or Mounting	Conduction	±20

(1) Integral Coldplate Concept (2) Vehicle Coldplate Concept

Table 5. Astrionic Equipment Component Typical Thermal Characteristics

Component	Typical Des Max Temp (°F)	Tolerable Max Op Temp (°F)	Tolerable Min Op Temp (°F)	Reference Temp Location	Reason for Temp Limit	Temperature Regulation Requirements
Typical Circuitry	+ 80	+130	- 65	Ambient	Rel, Performance	N/A
Power Transistor	+200	+275	- 65	Junction	Reliability	N/A
Signal Transistors	+160	+180	- 65	Case	Reliability	N/A
Integrated Circuits	+190 +100	+265 +180	- 40 - 40	Junction Case	Reliability	N/A
TWT - Low Power	+140	+160	- 65	Mtg Surface	Reliability	N/A
TWT - High Power	+180	+200	- 65	Mtg Surface	Reliability	N/A
Gyros (IMU)	+180	+200	- 65	Case	Performance	± 1°F
Gyros (Sensor)	+180	+200	- 65	Case	Performance	±10°F
Battery Cells	+ 80	+110	- 50	Cell	Regulation	±20°F
Accelerometer (Dry)	+200	+250	- 65	Internal	Performance	±10°F
Silicon Diodes & SCR	+ 80	+130	- 65	Case	Reliability	N/A
IR Detectors	-100	0	-150	Detector Surface	Signal to Noise	±20°F
Optics	+160	+160	- 65	Optic Bulk	Distortion, Focus	±20°F
Photomultiplier	+100 - 40	+120 +130	- 65 -110	Surface Ambient	Reliability Signal to Noise	N/A N/A
Vidicon Tube	+100 + 80	+160 +105	- 65 + 40	Optic Surface Face Plate	Signal to Noise	N/A
Capacitors (Tantalums)	+140	+200	- 65	Case	Reliability	N/A
Resistor	+250	+300	- 65	Case	Reliability	N/A
Motors	+250	+300	- 65	Windings	Reliability	N/A
CRT	+160	+200	0	Tube Body	Reliability	N/A
Crystal Oscillator	+160	+190	- 65	Crystal	Stability	±.01°F
Lasers	+220	+250	- 40	Laser Surface	Reliability	N/A

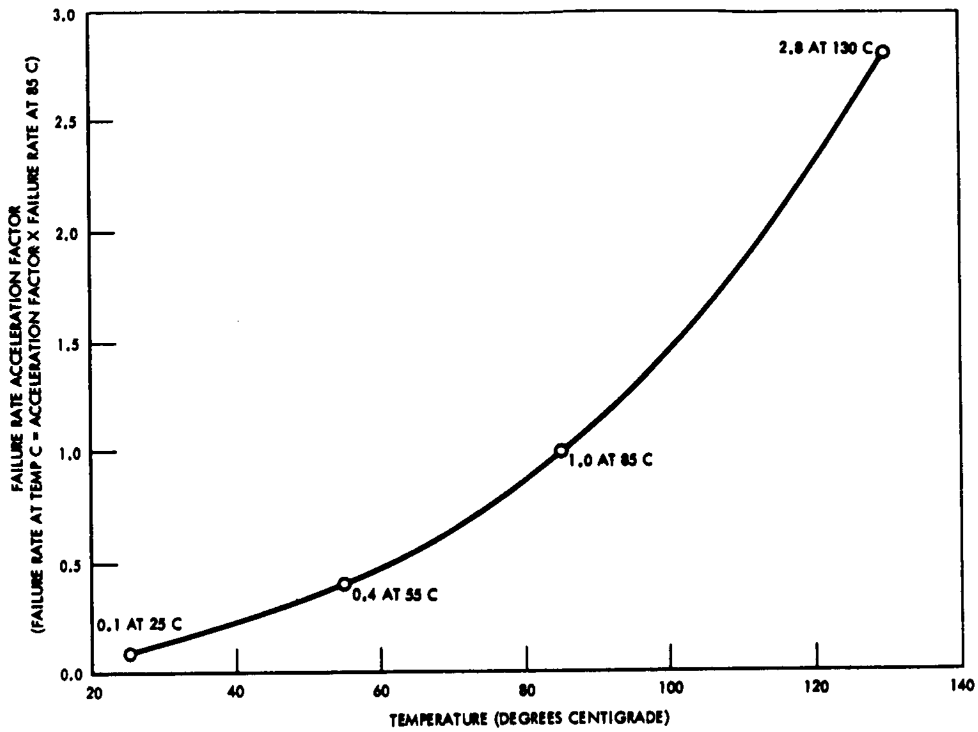
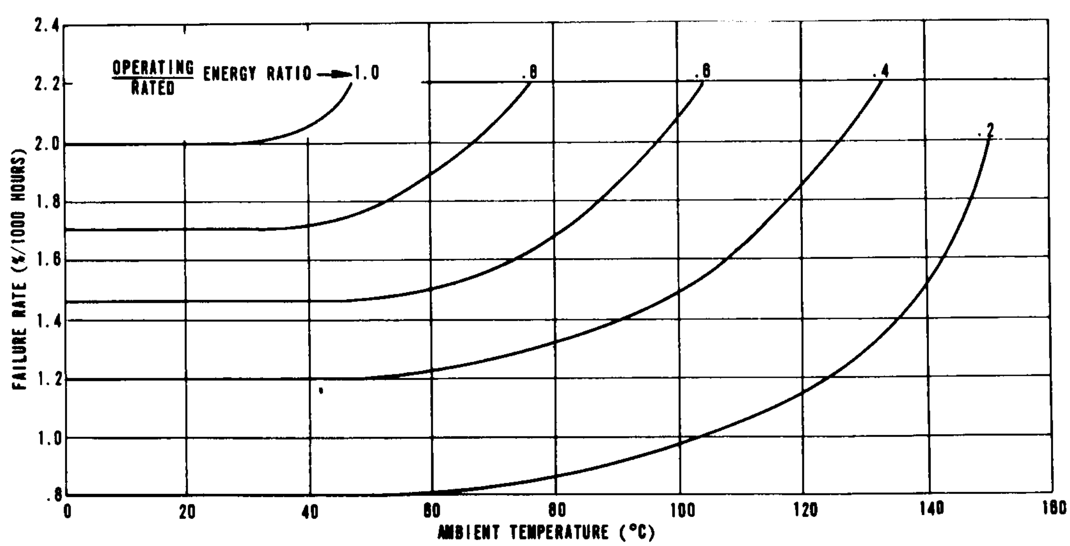


Figure 14. Integrated Circuit Performance Versus Temperature



NOTES: LEVEL OF CONTROL 2.
DATA SOURCE III-B, 1962

Figure 15. Failure Rates For Microwave Diodies, Mixer Application

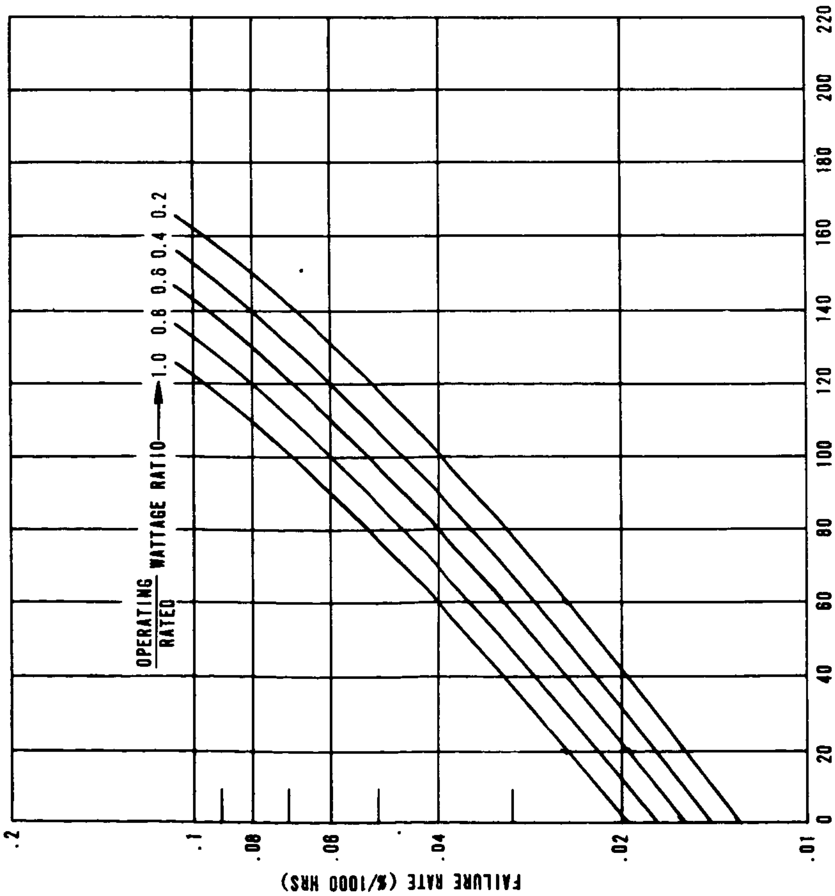


Figure 16. Failure Rates for MIL-R-10509D Film Resistors

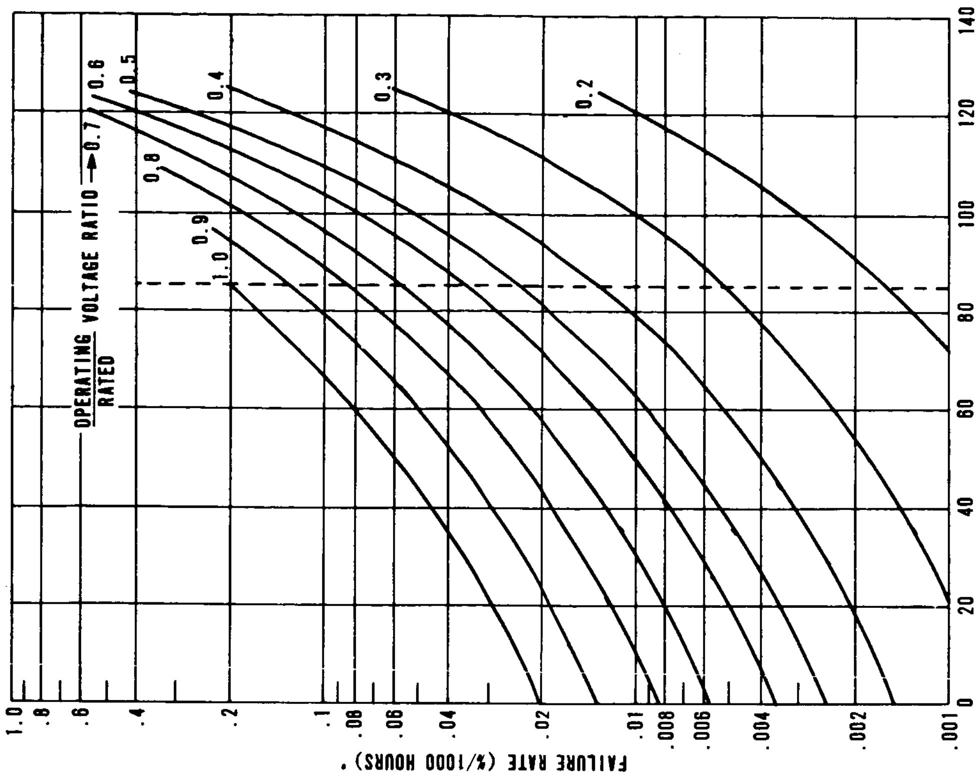


Figure 17. Failure Rates For MIL-C-39658 Wet Slug Tantalum Capacitor

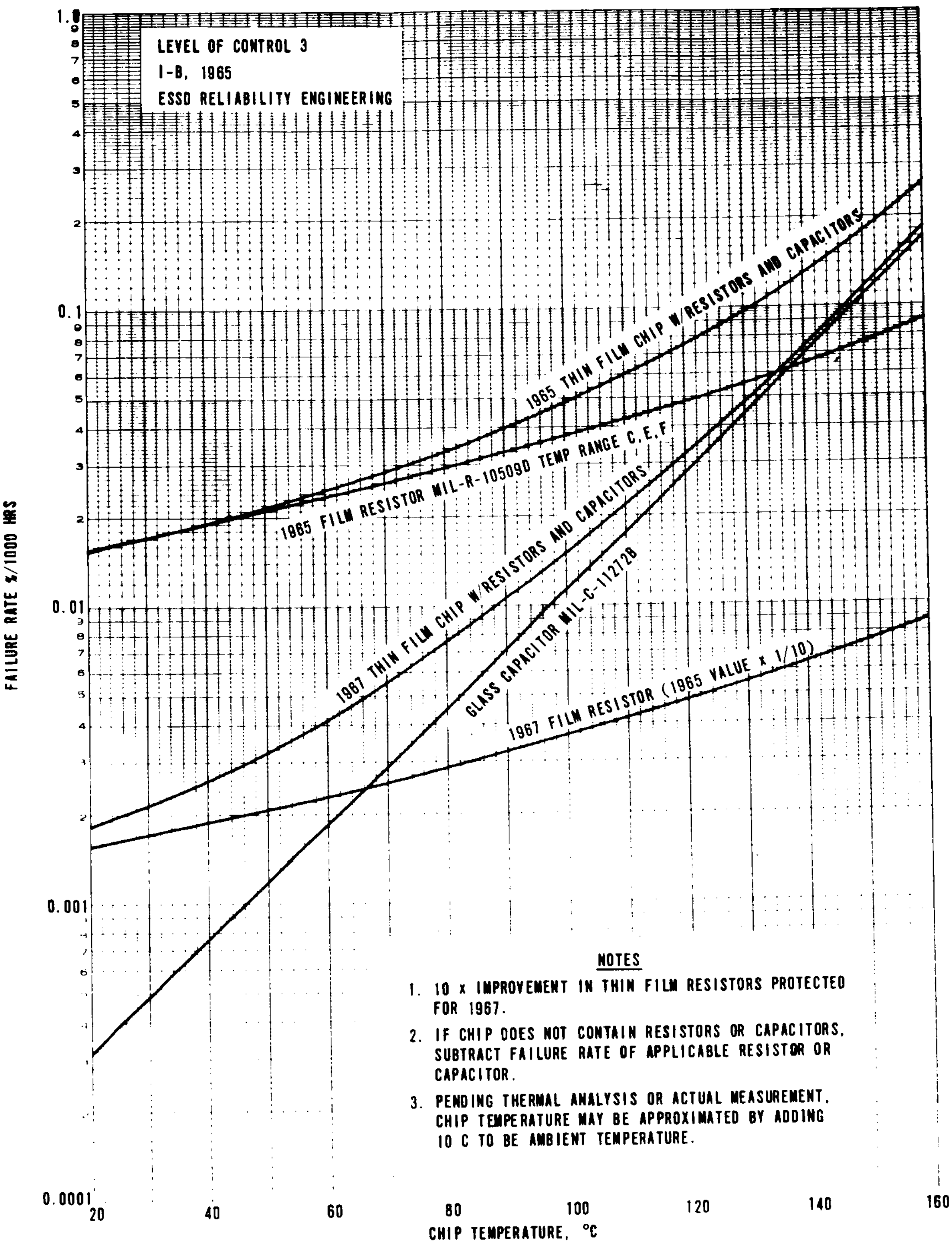


Figure 18. Thin Film Chip Failure Rates

Recent thermal studies using infrared microradiometer techniques (Reference 8) have shown the silicon semiconductor devices which are operating at supposedly safe "junction" temperatures of 150 to 200°C have, in reality, hot spots which are considerably higher, about 300 to 350°C. These studies suggest that with improved thermal design and more uniform distribution of the junctions within the silicon chip, a semiconductor device can be made to operate at higher temperatures with no decrease in reliability. Hot spots are dependent upon the circuit application of the device and upon micro-voids in the cement bond between the silicon chip and the metal case heat sink. As an example of the influence of circuit application,

Table 6. Failure Rates for Thin Film Chips and Ceramic Printed Circuits

TYPE CIRCUIT	STRESS	LEVEL OF CONTROL	FAILURE RATE	DATA SOURCE AND YEAR
Thin film chip with both resistors and capacitors *	50 C	3	.022	I-B, 1964 (ESSD)
Thin film chip with capacitors only *	50 C	3	.0012	I-B, 1964 (ESSD)
Thin film chip with resistors only *	50 C	3	.021	I-B, 1964 (ESSD)
Ceramic printed circuits with screened conductors	D	3	.0001	I-B, 1964 (NSD)
Each screened resistor			.005	

* See Figure 18

Table 7. Failure Rates for Silicon Integrated Circuits

IID NUMBER	TYPE CIRCUIT	STRESS °C AMBIENT	FAILURE RATE*	DATA SOURCE AND YEAR **
447-0010	Flip Flop	25	.001	I-B, 1965
-0011	Triple Nand Gate		.00075	
-0012	Clocked Input Nand		.00075	
-0013	Clocked Dual Nand		.0009	
-0014	Input Network		.001	
-0015	Output Driver		.0011	
-0017	Write Switch		.0009	
-0018	Matrix Switch		.0014	
-0019	Low-Level Switch		.0007	
-0020	Read Preamplifier		.001	
-0021	Level Detector		.001	
-0024	Gen Purpose Amplifier		.001	
-0025	Gen Purpose Amplifier		.001	
-0026	Gen Purpose Amplifier		.001	
-0028	Demod Chopper		.0008	
-0029	Driver Switch		.0012	
-0030	Power Switch		.002	
-0034	GPA-2A		.001	
-0035	Unclocked Dual Nand		.001	
-0040	Triple Nand Nonresistive		.008	
-0043	Input Network		.001	
-0056	Gen Purpose Amplifier		.001	
-0072	05 MV		.0008	
-0096	Level Detector		.0011	
447-0097	Write Switch	25	.0007	I-B, 1965

* Criteria of failure assumes:

1. Parts have met all electrical, mechanical and qualification requirements excluding burn-in.
2. System design utilizes the three year end points.

** Document T4-1732.1/33 Estimates of Inherent Failure Rates for WS133B Electronic Component Parts, Autonetics, Jan 1965

Company/Facility	Person Seen	Date
ITT Federal Laboratories Nutley, New Jersey	L. Pollack W. Glomb R. Lewthwaite R. Harrison J. Marley	9-23-66
RCA Astro-Electronics Div. (Space Center) Princeton, New Jersey	D. Brennen R. Scott P. Trusello A. Garfinkel	9-26-66
Westinghouse Baltimore, Maryland	C. I. Denton G. Perleberg R. Borrello P. Kiefer H. Kraus	9-27-66
Litton Industries Data Systems Division Van Nuys, California	E. Baker	

Other organizations, primarily in the west coast area, will be visited during the current quarter and the data/opinions obtained will be used to supplement information obtained to date.

Each organization visited were basically asked the following questions:

(1) What are your experiences regarding trends in heat loads required to perform particular functions?

(2) What are your important design criteria with respect to critical temperature limits and regulation requirements?

(3) What heat transfer design techniques do you employ or anticipate employing with higher heat densities for removing heat from where it is generated to the package envelope?

(4) What are your experiences on integral coldplate versus vehicle coldplate cooling concepts?

a silicon power transistor operating at 50 watts consisting of 5 amps at 10 volts behaves thermally quite differently from the same device operated at 2 amps at 25 volts. These effects cause the greatest variation (and error) in the internal thermal resistance or the "°C/watt" approach to desired junction temperatures.

Astrionic Equipment Internal Design State-of-the Art

A number of organizations familiar with the design of astrionic equipment was visited during the past quarter by H. Kamei and F. B. Iles of Autonetics. The purpose of these visits was to assess the state-of-the-art of astrionic package thermal design. As explained in the first quarterly report (Ref. 1), micropackaging (packaging of microelectronics) plays a significant role in thermal design, and therefore micropackaging efforts were also assessed.

Persons and Companies visited were the following:

Company/Facility	Person Seen	Date
Texas Instruments Corporate Research and Engineering Dallas, Texas	D. Peterman	9-14-66
Sylvania Electronic Systems	J. J. Staller I. Greenburg M. Knepp H. Lake R. Deverde M. Berberian	9-20-66
Massachusetts Inst. of Tech. Instrumentation Laboratory	K. Fertig F. Petkunas A. Hollander	9-21-66
IBM Electronic Systems Center Owego, New York	F. J. Price M. C. Panaro C. Blivin R. A. Monroe M. Rauhe	9-22-66

The following general conclusions have been reached. These preliminary conclusions support the instrument packaging constraints/requirements previously postulated (see Reference 1, p. 26).

1. Heat Load. Astrionic equipment overall heat load will stay relatively constant. The large power users are least miniaturizable in power. Heat load reduction obtained by microminiaturization will be countered by increased equipment functions and complexity. Detailed heat load data for specific equipments or functions have been included in Table 4 and 5.

2. Package Internal Heat Transfer Techniques. Essentially, all astrionic packages use conduction as the principal mode of heat transfer inside equipment packages. Nothing new was encountered. Various people had various ideas on: (a) materials for reducing interface contact resistance such as greases, RTV and soft metals, (b) module hold-down features such as the "wedge", metal-to-metal fasteners, and springs, (c) means for promoting conduction across circuit boards such as using alumina boards, using metal backup plates, and variations of the "copper rail," and (d) heat conduction encapsulant materials such as dry alumina granules, alumina and beryllia-filled plastics, silica-filled plastics, etc.

   Most companies have not experienced sufficiently high heat loads to warrant forced convection or ebullient cooling inside packages. (Internal forced convection, where used, has been for obtaining temperature regulation).

3. Temperature Regulation. Temperature regulation, where used, is predominantly heater controls.

4. Integral Versus Vehicle Coldplate. All astrionic equipment companies with experience with vehicle coldplates are convinced that vehicle coldplate not the best method. Invariably, weight can be saved, coolant utilization can be improved, and operating temperature can be improved with resulting improvement in reliability by the integral coldplate method. IBM's Saturn V computer and data adapter is now integral coldplate. Companies using the vehicle coldplate concept clearly point out that it is a requirement imposed on them.

5. Critical Temperature. Most designs are governed by equipment reliability; the lower the device and component temperature, the higher the reliability. A maximum junction temperature of 125 degrees centigrade is commonly used for silicon semiconductor devices. Gyros operate fairly high, about 170 to 180 degrees Fahrenheit, and require the closest temperature regulation. Batteries have the most severe temperature requirement and are the weakest link in the thermal design. A range of 10 to 35 degrees centigrade is tolerable, but 15 to 25 degrees centigrade is preferred.

   The most frequently mentioned temperature limit was the junction temperature of silicon semiconductor devices (transistor, diode, integrated circuit). Table 8 gives the maximum temperature used as a design criteria by a number of organizations surveyed.

Table 8. Silicon Devices Max Temperature Design Criteria

Organization	Criteria Temp (°C)	Temp Reference Location	Remarks
Autonetics	125	Junction	Minuteman criteria
Honeywell	125	Junction	
IBM	125	Junction	Normal; 100°C desired; 75°C very conservative.
ITT-FL	125	Junction	Rated; 110°C used.
Litton - DSD	80	Case	Low power devices.
MIT Inst Lab	105	Junction	Used for conservative design.
RCA - Space	150	Junction	
Sylvania	100	Junction	Design objective; 175°C abs. max.
TI	250	Junction	Possible with proper thermal design.
Westinghouse	125	Junction	Normal; 85°C very conservative.

2.2 COMPONENT AND SUBSYSTEM REVEIW AND EVALUATION

The survey of current and advanced state-of-the art in astrionic equipment development and associated thermal control and packaging design techniques, and thermal control components has been the major effort within this task. The survey effort has consisted of visits or contacts with important equipment manufacturers. This survey effort is a continuing one and will extend into the next quarter.

In conjunction with the survey, a review of some of the important components of a thermal control system has been made. These are discussed in the following paragraphs. The review and evaluation effort is to be continued and will include all the significant components and subsystems.

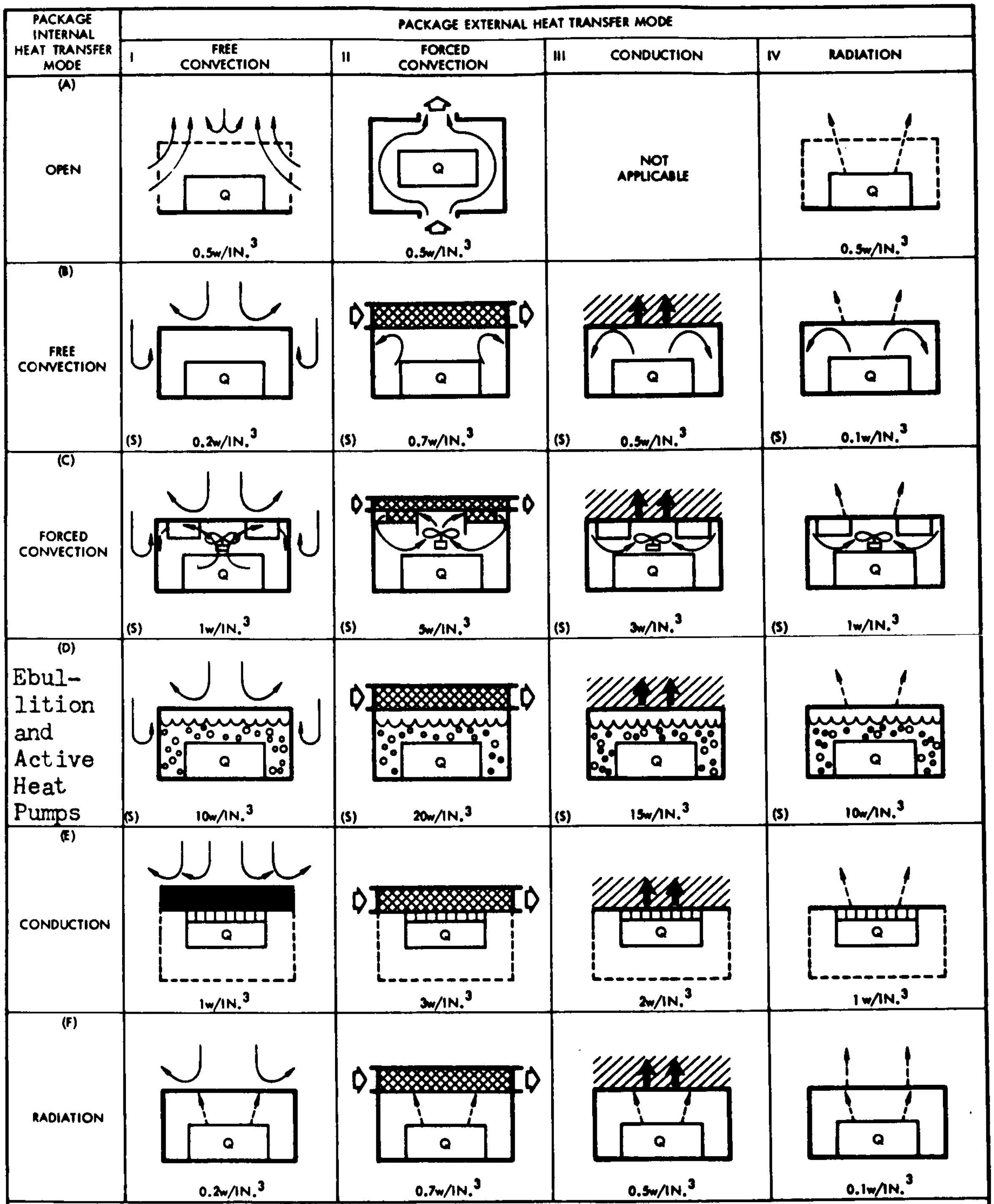
2.2.1 Astrionic Equipment Package Thermal Design

Heat is transferred from and within astrionic packages by convection, conduction, radiation, or a combination of these modes; however, one mode usually predominates. Convection may be natural (free), forced, or accompanied by a change of phase such as boiling. Figure 19 shows all feasible instrument cooling concepts, categorized by the predominant heat transfer mode within and external to the package. Figure 19 also indicates the approximate maximum heat-removal capability for a temperature difference of 40°C. In general, more efficient heat removal can be accomplished at the penalty of increased weight and complexity, e.g., forced convection with the requirement of pressure-tight enclosures. Free convection is not applicable at zero-g condition or in the vacuum of space; however, this mode is shown on Figure 19 for comparison. The following paragraphs are concerned with package internal heat transfer.

Internal Heat Transfer Devices/Techniques

Various heat transfer techniques are utilized within astrionic packages. These include the following:

(a) Conductive Materials - Internal electronic package heat transfer by conduction has become the most important method and will become increasingly dominant as microminiaturization continues because: (1) conduction heat transfer is the most predictable and reliable (meaningful tests can be conducted at laboratory conditions), (2) miniaturization in size has shortened conduction paths, (3) significantly smaller package heat loads with higher heat densities exist, (4) heat conduction paths can be tailored to the heat source temperature and wattage, and (5) the pronounced trend to integral coldplates



NOTES:

(1) MAXIMUM HEAT TRANSFER CAPABILITIES AT $\Delta T = 40°C$ INDICATED.
(2) S INDICATES PRESSURE-TIGHT ENCLOSURE REQUIRED.
(3) * INDICATES NOT APPLICABLE AT ZERO g AND/OR SPACE VACUUM ENVIRONMENTS.
(4) CONVECTION ——→, CONDUCTION ➡, RADIATION — — —→, Coolant Flow ⇨

Figure 19. Electronic Instrument Cooling Concepts

is evident. It is expected that this method will be used in nearly all electronic equipment in the near future, particularly for predominantly digital circuitry equipment.

To the extent practical, conduction is relied upon through well-defined, high heat conductive (metallic) paths. Steady-state heat transfer by conduction is governed by $\dot{q} = KA\,(\Delta T)/l$. Since maximum heat transfer is desired at minimum ΔT, it is obvious that a large K (high conductivity materials), a large A (cross-sectional area) and a small l (conduction length) are desired. Tables for the conductivities of various materials commonly used in manufacture of astrionic equipment are readily available in reference handbooks and design guides. Metals such as copper, aluminum, and silver have highest conductivity, followed by alloys such as those for aluminum and magnesium. Since weight is of great concern to astrionic equipment, lightweight metals such as magnesium, beryllium, aluminum, and maglithium are commonly used. Most non-metals are poor conductors; beryllia (BeO) and alumina (Al_2O_3) are significant exceptions which are used where their properties of electrical insulation are required.

(b) Encapsulants - Many electronic equipment, primarily those fabricated by the "welded module" approach, are encapsulated or "potted" for environmental protection. No ideal encapsulant is available and a compromise exists with potted electronics between weight, repairability, and heat transfer (conductivity). Where thermal requirements are negligible and weight is the main factor, potting is accomplished by semi-rigid foams, or by encapsulants which have low density fillers such as "microballoons." Where heat transfer is required, fillers for encapsulants are usually high conductivity materials such as anodized aluminim particles and alumina or beryllia granules. A recent innovation has been the use of dry alumina granules in hermetically sealed cans without the plastic binder. Commonly used binders include epoxies and silicone rubber compounds.

(c) Component Fins - Component fins (erroneously called heat sinks) are commercially available which promotes convective heat transfer from devices such as transistors by acting as extended surfaces. Figure 20 shows sketches of typical transistor "heat sinks."

(d) Module Retainers - The primary problem associated with conduction inside the packages (as well as outside the package) is contact thermal resistance. If internal package modules, such as the heat-conducting etched-circuit-board assemblies, are permanently or semi-permanently cemented into place, contact resistance can be minimized. This approach, however, would make repairs impractical and throw-away costs excessive. A unique solution is offered by the "module-lock" concept used successfully in the Minuteman program. The "module-lock" concept is depicted by Figure 21. Metal spring-type module retainers are also depicted by Figure 21. These are commercially available, fabricated in beryllium copper, phosphor bronze, or "Danalloy," a

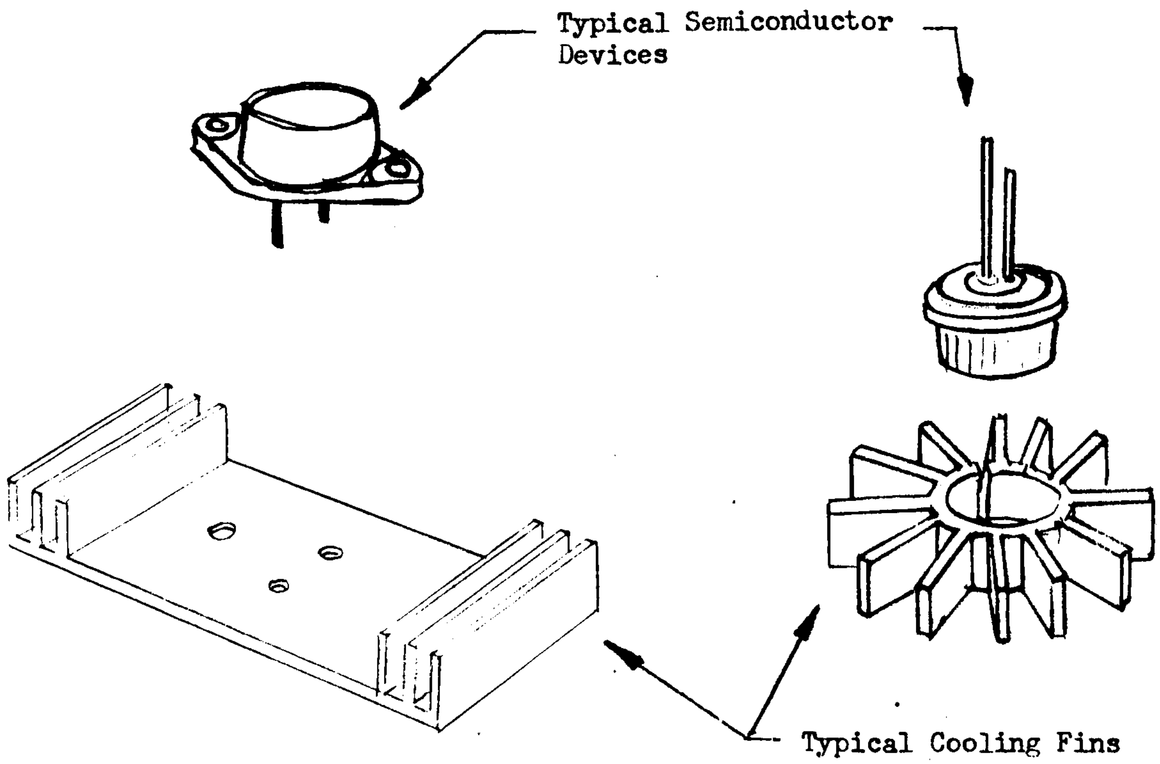


Figure 20. Typical Transistor Cooling Fins

silver heat-treatable alloy of very high thermal conductivity. The function of the module lock and spring module retainers is to apply pressure between the module and the coldplate to promote heat transfer, i.e., reduce contact thermal resistance. Usually the contact areas are wiped with silicone grease not only to reduce the contact thermal resistance, but also to reduce sliding friction.

(e) <u>Module Heat Shunts</u> - Modules which use epoxy glass circuit boards, glass substrates, and other low heat conducting materials are frequently severely limited in heat density without some form of heat shunting. One common method is metal backup or a metal sandwich construction. Aluminum and magnesium are commonly used. The metal backup concept is illustrated, as applied to CPC's (ceramic printed circuits), in Figure 22. Another method is the "copper rail" where a copper or aluminum bar is cemented onto the circuit board and heat dissipating active components such as integrated circuits are cemented to the rail. Another technique often used isto leave, wherever possible, thecopper clad material in circuit boards to act as thermal conduction paths. The clad is often thickened by dip soldering.

(f) <u>Convection Techniques</u> - Convection techniques involve the circulation of a fluid, either forced or naturally, in a sealed container. For orbiting space vehicle applications, natural convection does not work and, therefore, a fan or pump is required. Because of

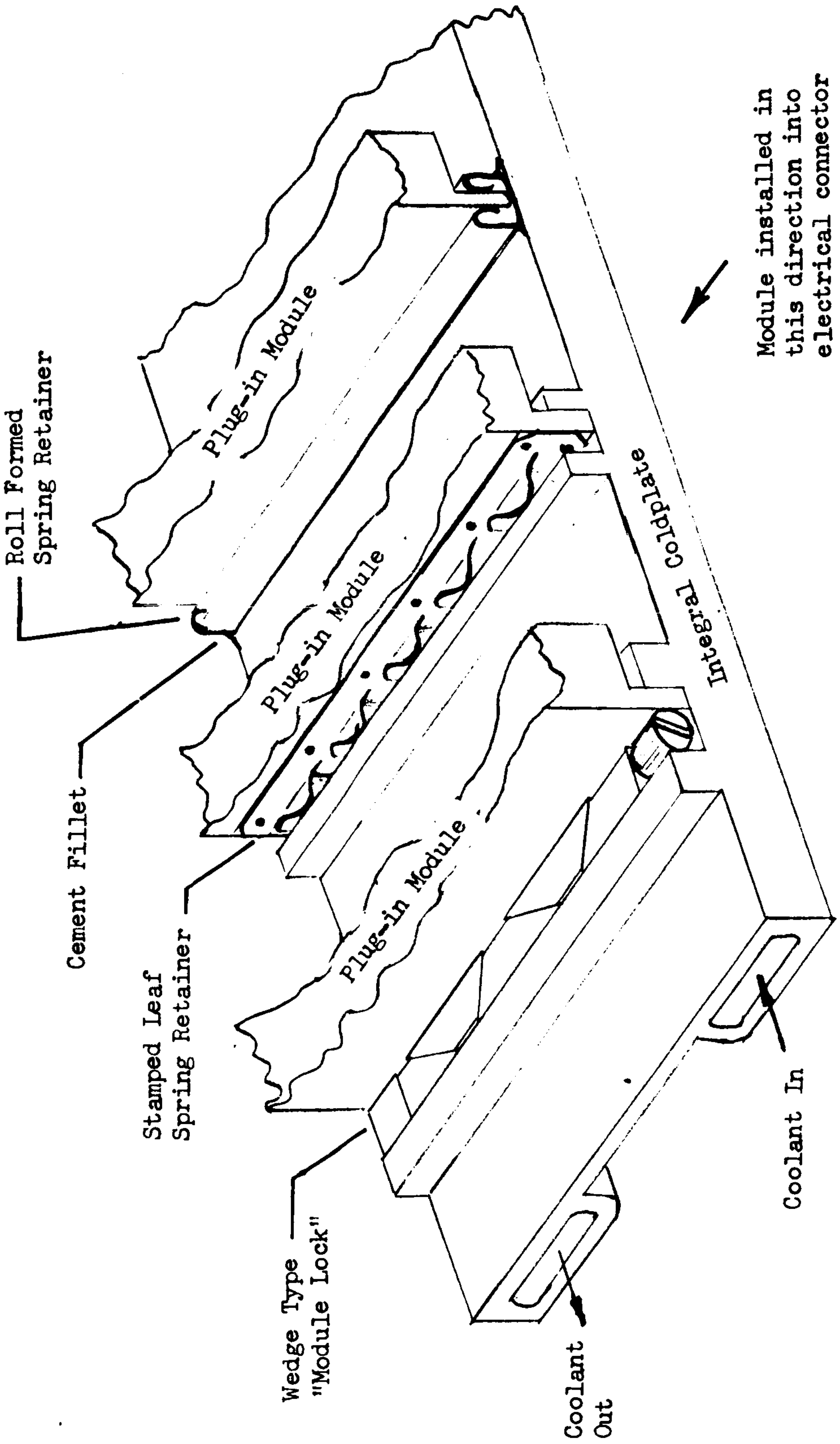
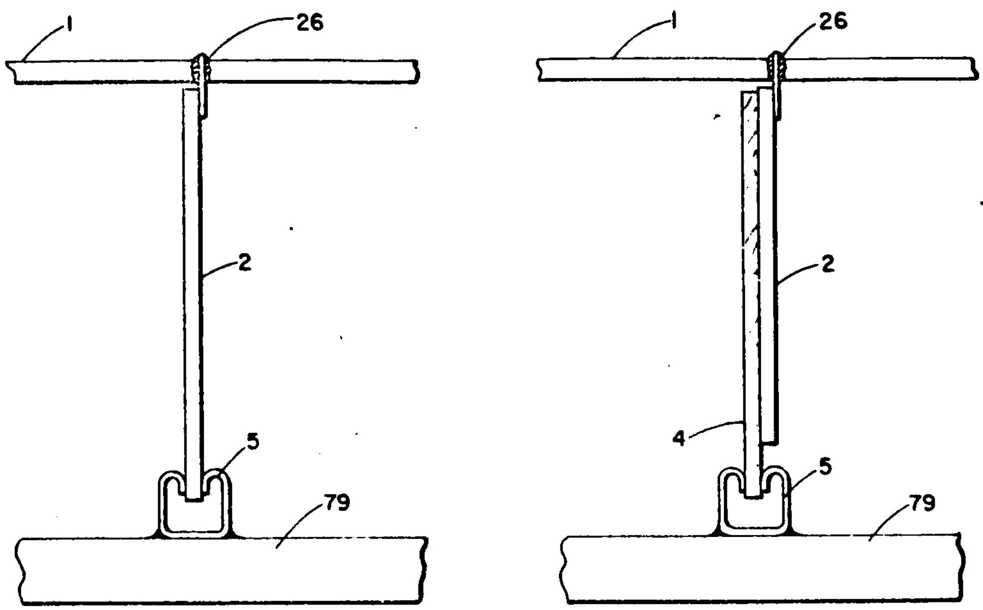
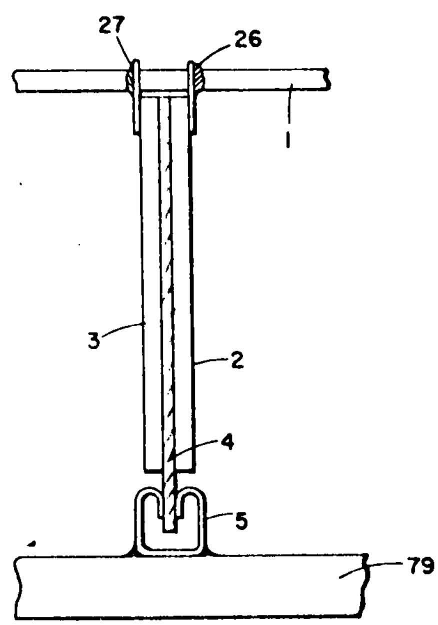


Figure 21. Typical Module Retainers





1 - Interconnecting Multilayer Board
2, 3 - Ceramic or Glass Epoxy Circuit Board
4 - Metal Back-up Plate (Heat Shunt)
5 - Module Retainer - Spring Type
26, 27 - Electrical Connection Lead
79 - Coldplate or Package Wall

Note: See U.S. Patent 3,268,772
Hiroshi Kamei, et al
August 23, 1966

Figure 22. Metal Back-up Plate Type Heat Shunts

the need for the fan or pump and the need for the hermetic or semi-hermetic sealed enclosure, convection techniques are not commonly used. There is, however, limited use with complex, precise electro-mechanic devices such as a gimballed stable platform. In these limited cases, however, the internal coolant loop is presently required primarily for ease in precise temperature control.

With increased heat densities resulting from microminiaturization trends, forced convection method will become more important for cooling application, perhaps as a supplement or backup for conduction cooling. Usually, the internal coolant is the initial charge of an inert dry gas. A leakproof package seal and a long-life blower are two major concerns.

The best seal is a hermetic seal, consisting of metal-to-metal or metal-to-glass fusions. For accessibility, however, semi-hermetic seals are required which include various gaskets such as the O-ring and molded seals such as Parker's gask-O-seal. Both the hermetic and semi-hermetic seal, if used, can (and must) be tested for their integrity. As an example, the Minuteman I guidance and control equipment, which is hermetically sealed in a one-sixteenth inch wall thickness magnesium enclosure containing about 10 cubic feet of helium at nominally 18 psia, is factory-tested so that the leakage will not result in a pressure loss of 2 psi over 8 years. A mass spectrograph-type helium leak detector is usually used. If the inert gas used is not helium, helium can be added as a tracer; 5 or 10 percent by volume is commonly used.

Compact and low-weight blowers are necessarily high-speed, high power and low-life devices. Typical life of high-speed ball-bearing blowers is 1000 hours, however high-speed blowers can last over 5000 hours with special grease-pack and quality control. Longer duration would necessitate hydrodynamic gas-bearing blowers. Self-lubricating gas-bearing blowers are used in the Minuteman I guidance package and in the Minuteman II stable platform package which have a three-year operating life requirement.

(g) <u>Ebullient Cooling</u> - Ebullient or direct liquid cooling is accomplished by submerging the hot components in an inert fluorochemical liquid. Heat is transferred from the hot component to the liquid by conduction and vaporization, and from the liquid to the package walls by convection and condensation. High heat dissipation factors are obtained, and therefore, this cooling method may have application for future compact electronic equipment. Studies and tests of a computer power supply cooling application are reported in Reference 9.

Major disadvantages for this method are the following:

(1) Effectiveness is limited in a zero-g environment.

(2) Sealed and pressure-tight enclosure is required.

(3) Fluid has high thermal expansion requiring expansion chambers, e.g., rubber diaphragm, metal bellows, and hollow balls.

(4) Repair and servicing limited to facilities having fill capabilities.

(5) Increased weight due to (2), (3) and the fluid itself.

(h) Heat Pumps - Heat pumps may be effectively utilized to pump heat from its point of origin at the active components to the walls of the package. The application of thermoelectric refrigeration for this purpose was described in the first progress report (Reference 1) and further discussions are included elsewhere in this report. The major disadvantage of the thermoelectric heat pump is the requirement for electrical power.

A heat pump method which does not require electrical power is the "heat pipe" (Reference 10) concept. The heat pipe is a unique heat transfer device which may be applicable where a large quantity of heat must be moved effectively and at low cost. The physical processes involved are vapor heat transfer and liquid capillary action. The pipe consists of a tube closed on both ends which has a capillary structure (wick) along its inside surface. The pipe is evacuated and a small amount of fluid is introduced which saturates the capillaries and has a significant vapor pressure at the desired operating temperature. The fluid is evaporated at the heat input end, condensed at the heat output end, and is returned to the evaporator (heat input end) by capillary attraction. The heat pipe has an effective "conductivity" many times higher than a metal conduction bar of the same size and would obviously have much less weight.

(i) Radiation - Radiation heat transfer of astrionic packages are presently used in some low heat dissipation equipment which are not amenable to conduction such as the Apollo Flight Director Display. Unless high temperature components become universally available, radiation heat transfer within the equipment packages will become decreasingly important.

Temperature Control Devices/Techniques

Astrionic equipment packages containing high temperature sensitivity modules and components require sophisticated control systems. The degree of temperature regulation required is a function of the required equipment performance and reliability. In general, equipments provide better performance and reliability with closer temperature regulation. Performance dependence on temperature regulation is most clearly illustrated by inertial stable platforms whose performance is directly related to performances of their inertial instruments (gyros and accelerometers). These inertial instruments have temperature sensitive drift (error) coefficients which require temperature control from ± 0.01 F to ± 10 F, depending on the application and the type of instrument.

Temperature regulation requirements are expected to remain unchanged in future astrionic equipment from current equipment requirements. For example, the reduced temperature coefficients of improved inertial instruments such as the case rotated, gas bearing free rotor gyros and dry electromechanical accelerometers will be balanced by demands for improved system performance due to longer operating duration. Also, in the case of semi-conductor devices in future microelectronics, the complex integrated circuits will demand temperature regulation similar to present day integrated circuits and discrete transistors. This is expected since all three - the transistor, the integrated circuit, and the complex integrated circuits, are silicon devices manufactured by basically the same techniques and processes.

Temperature regulation methods applicable within packages are not basically different from methods applicable outside of the packages. They include heater control systems, heating and cooling control systems, coolant flow control systems, and storage systems.

Heater Control Systems - The heater control system is the most common means for maintaining close temperature control. The essential elements of this system are the sensor, the electronic controller, and the heater. In the simplest form, the sensor and controller is a thermostat (good for ± few degrees F) or a thermostat with a solid state or mechanical relay (good to ± 1/4 F). Precise temperature control (better than ± 0.1 F) requires an electronic controller. The method is quite versatile in that various modules can be controlled to their individually desired control point.

In a heater system, the equipment is over-cooled and make-up or "bias" heat is applied to maintain a "constant" heat load. In more complex systems, an internal coolant loop is provided to facilitate temperature control where the heater rejects heat to the circulating coolant.

Reversible Heating/Cooling Control Systems - Reversible heating/cooling control systems are frequently employed where electrical power is at a premium. Such a system is typified by the thermoelectric temperature controller described elsewhere in this quarterly report. These systems have the advantages of: (1) minimizing peak heater power, (2) operating at lower control temperature, allowing higher reliability of the equipment, and (3) decreasing reaction time from a cold-start condition. The first two advantages are illustrated by Figure 23. The equipment temperature variation without temperature control could be due to changing equipment heat load, changing environment, changing heat sink (e.g., coolant from the vehicle ECS) tempera-

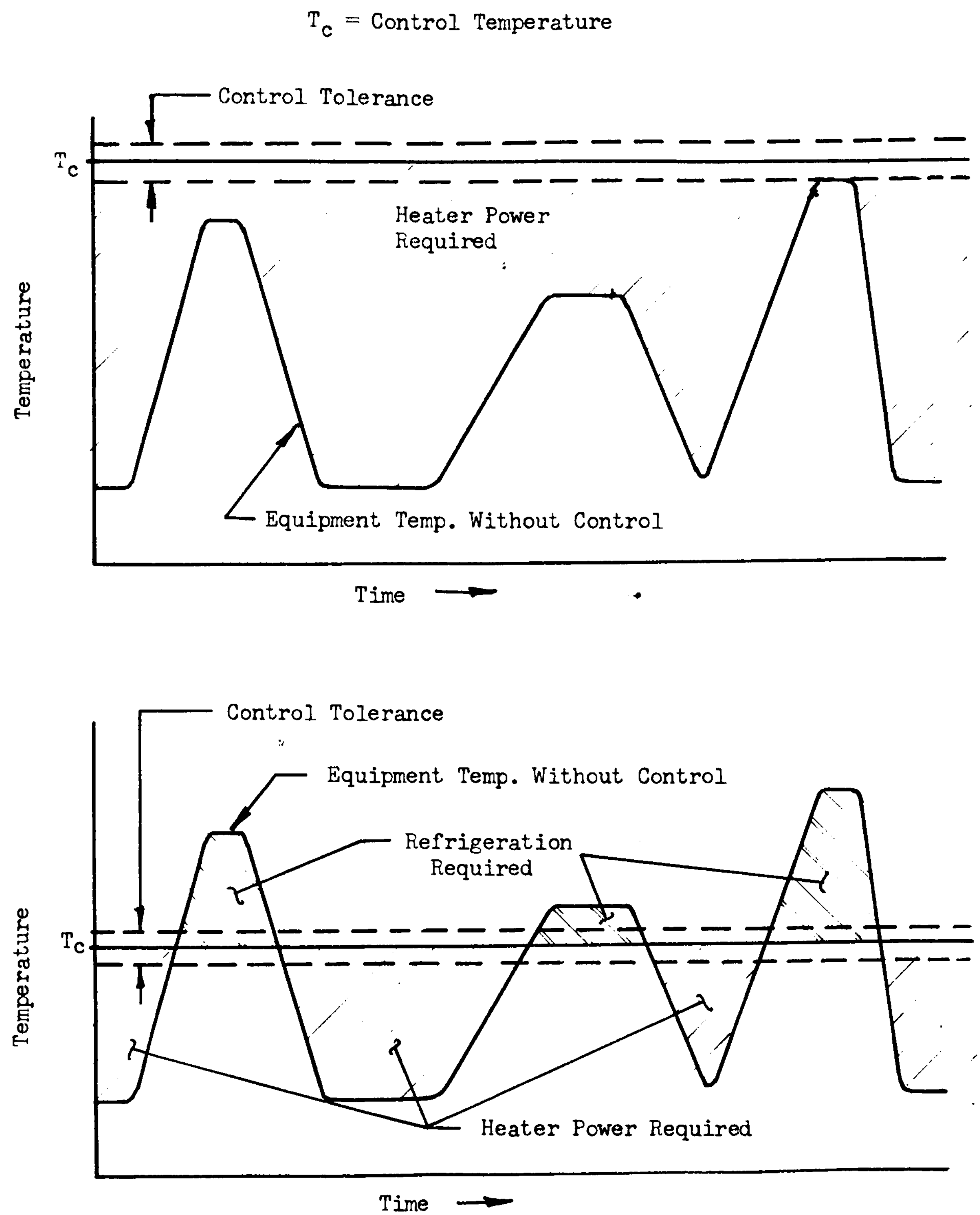


Figure 23. Comparison of Heater Control System with Heater/Refrigeration Control System

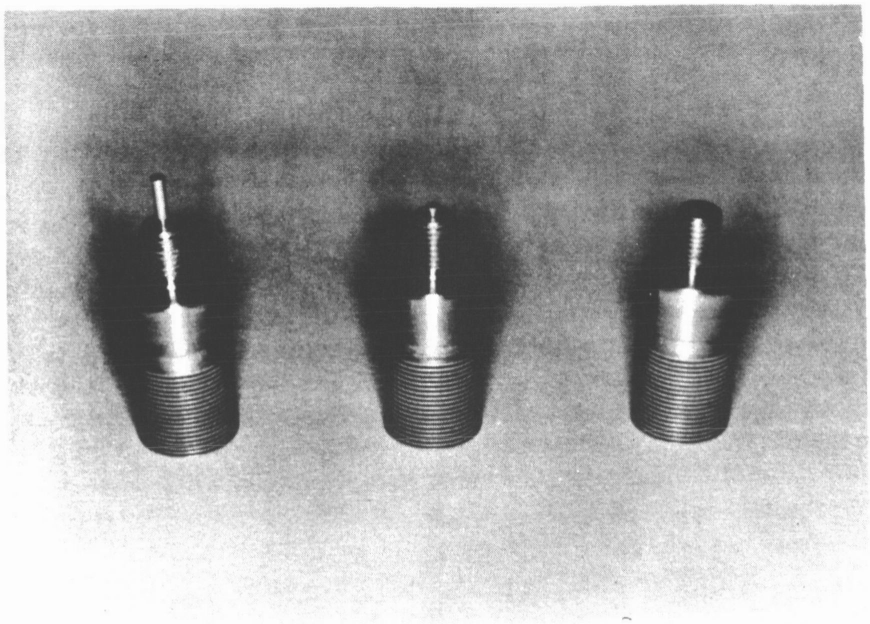
tures, or their combinations. The temperature control tolerance can be made as small as for the heater system; a tolerance of ± 0.01 F is possible with an electronic controller and thermistor bridge sensor.

As an alternative to operating at lower control temperature, the reversible heating/cooling method can be used as a means of tolerating a wider heat sink temperature range. For example, a vehicle may have a number of electronic packages which can tolerate liquid coolant from the vehicle ECS in the range of -50 F to +150 F, and a few packages which require coolant in the range of 0 F to 100 F. Instead of providing coolant at the narrower temperature range to all the packages, the wider range can be provided to all the packages. Those packages requiring the narrower temperature range will have incorporated within them the provisions for reversible heating/cooling. In effect, this is a coarse temperature control method where the control band is the narrower temperature range, that is, heating is provided when the coolant drops below 0 F and cooling (local refrigeration) is provided when the supplied coolant exceeds 100 F.

<u>Flow Control of Coolant System</u> - Thermally actuated coolant system flow control valves, i.e., "thermovalves", provide a non-electrical means of attaining proportional temperature control of equipment packages. Proportional temperature control is maintained by varying the flow of coolant to the package heat exchanger by means of a mechanical valve which is located in the coolant line. The valve position is varied by the action of a thermal actuator through a mechanical linking device.

The thermal actuator usually consists of a cylinder of temperature sensitive wax with a high coefficient of expansion impregnated with highly conductive metal particles. The metal particles are utilized to minimize the thermal lag in the control cirucit, thus enabling a closer control of the recirculated gas temperature. Figure 24 shows a commercially available thermal actuator in three stages of operation. The design criteria are included to show the capabilities of this type of actuator. Similarly, the valve actuator could be a bi-metallic element, a bulb containing a 2-phase fluid, or an electric motor controlled by an electronic position servo.

The thermo-valve can be used in various configurations of environmental control systems. Figure 25 is a typical application of the device. In this application the thermo-valve is used to regulate the coolant flow rate by sensing the temperature of recirculated gas as it leaves the heat exchanger inside the equipment package. The thermo-valve is pre-set at the proper control temperature at installation under simulated operating conditions. The thermal actuator exhibits no deterioration with age and thus allows long term temperature control stability which makes it ideal for space systems. A small amount of hysteresis is encountered in the operation of the thermo-valve due to



Typical Thermal Actuator Characteristics

Weight - 1 ounce
Output force - 20 pounds
Ambient temperature range - -65°F to +250°F
Temperature control range - -20°F to +250°F
Linear between 30°F and 150°F
Rate of travel - .015" per degree F, or less when specified
Time constant (typical) - Starting at 50°F, immerse in fluid at 100°F.
Time to reach 60% stroke, 12 seconds.

Acknowledgement:
Pyrodyne, Inc.
11876 Wilshire Blvd.
Los Angeles 25, California

Figure 24. Commerical Thermal Actuator Eutectic Wax Type

friction and tolerances inherent in mechanical devices. Temperature control of the recirculated gas can be maintained within a ± 2 F range about the set point without difficulty for ambient temperature in the range from -65 to +160 F. Finer temperature control requires local temperature controller of the heater type which can operate within the coarse temperature control provided by the flow control system.

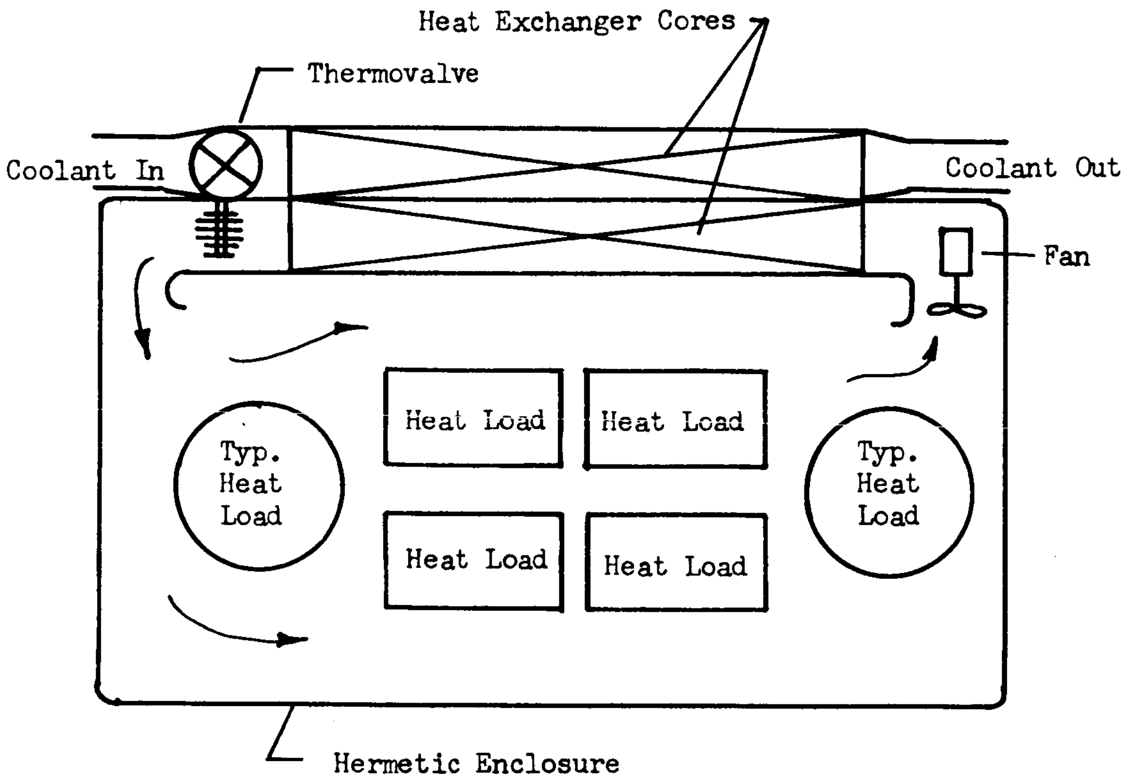


Figure 25. Schematic of Typical Flow Control Temperature Regulation System

Another form of flow control temperature regulation is internal circulating blower (or pump) speed control. Temperature control of heat generating components is accomplished by blower speed control since convective heat transfer is a function of the flow (velocity) of the recirculating coolant relative to the heat dissipating component.

<u>Storage Systems</u> - Storage systems can be used inside packages to provide thermal control in the same manner that they can be utilized for spacecraft thermal controls. Temperature control is obtained by utilizing the heat of fusion of various materials to absorb the excess heat. Change of state storage systems are described in References 11 and 12. These references adequately describe the storage system.

2.2.2 Expendable Cooling Methods

For relatively short duration cooling operation, the expendable heat sink provides a reasonably simple, lightweight system for continuous or intermittent operation, depending upon the application. The present design of the thermal control system for the instrument unit utilizes the expendable heat sink as the primary cooling method because of the short duration operation. For a longer duration operation, the expendable heat sink may be used in conjunction with a non-expendable heat sink such as a space radiator. For this type of application, the expendable heat sink may be designed to meet peak loads (spikes) and for other possible situations when the non-expendable heat sink is non-operative. One possible situation would be an emergency due to failure of the primary heat sink such as a space radiator. Another possible advantage in utilizing the expendable heat sink as an integral part of the thermal control system, is the convenience in expanding the cooler capacity in terms of heat load and/or operating time without major system changes or modifications.

There are a number of possible expendable cooling alternatives and these are illustrated in Figure 26. The selection of a particular method will depend upon the availability and source of coolant, the design requirements of the major components and the overall weight, power and volume savings. Stored or transported, expendable (water) method is used in the present instrument unit thermal control system. An example of a possible utilization of generated fluid (boil-off) and the residual fuel is that of hydrogen from the Saturn S-IVB stage. This investigation is discussed in detail in a later section.

The general area of applicability of expendable cooling methods is briefly discussed in a later section on heat sinks.

Expendable Coolant Sources

Stored or Transported Expendables - Because of the weight and volume penalty, only those fluids or solids that have high heat of vaporization would be considered suitable as stored expendable. The design of the storage container, expulsion method and the heat exchanger will also influence the selection of the coolant.

In Reference 11, pages 39 through 44 give a list of possible expendable coolants and their thermal properties. From those listed, only water (h_v = 1000 Btu/lb), ammonia (h_v = 570 Btu/lb), and ammonia-water solutions (h_v = 950 Btu/lb) have sufficient heat of vaporization to be considered as liquid transportable expendables. Formic acid (h_v = 560 Btu/lb) and acetimide (h_v = 560 Btu/lb) are subliming solids which can be considered for special applications. Utilization of the evaporating liquids follow existing technology. To date, the handling of subliming solids has been neglected. Though no particular dif-

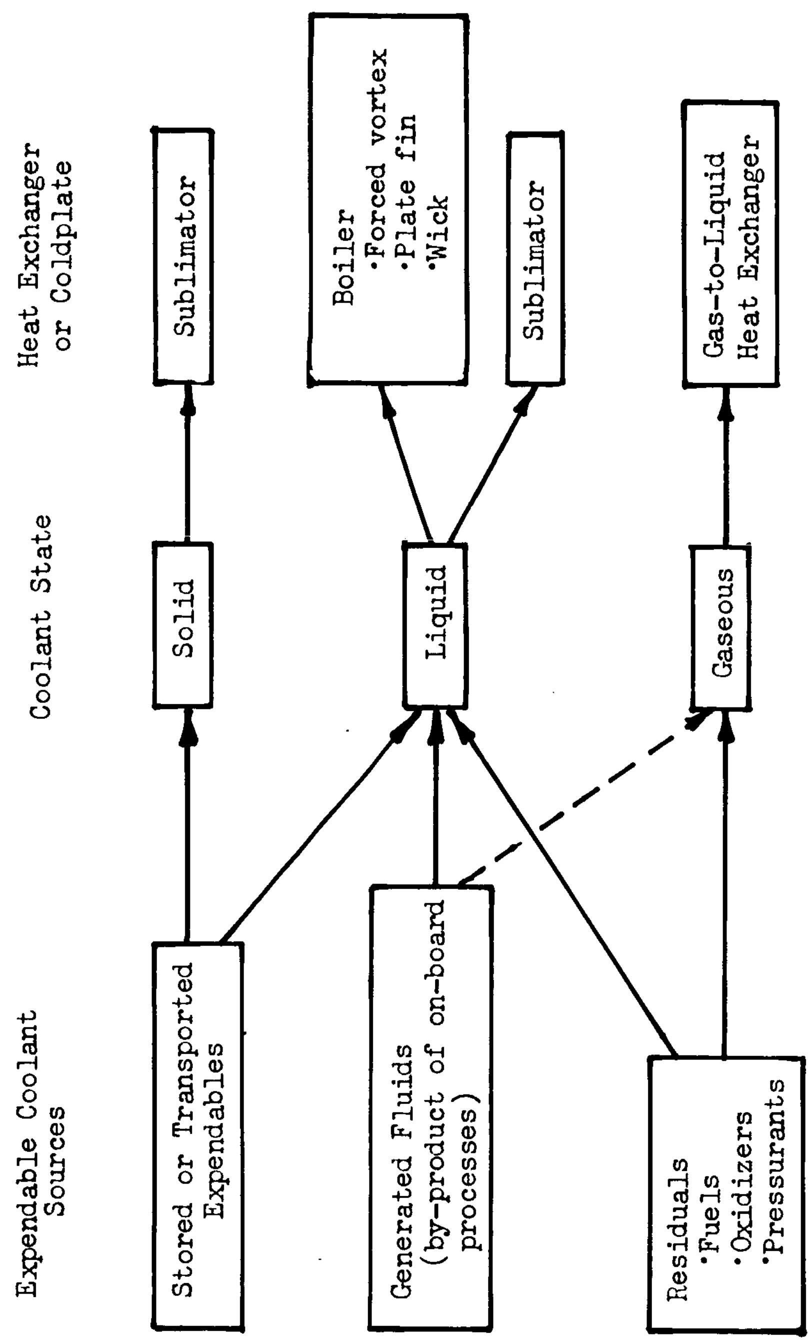


Figure 26. Expendable Cooling Alternatives

ficulties would be expected, some experimental work on the interface between the source and the subliming solid is required to establish the interface temperature drop.

Generated Fluids - Equipment and personnel from other functioning units of a space vehicle may generated fluids which can be useful for cooling. Water with differing amounts of contaminants may come from fuel cells, humidity condensers, wash water, and urine. Carbon dioxide also may be produced in appreciable quantities but the use of it in cooling is not practical.

The utilization of the generated fluid will depend upon a number of factors which includes: (1) the availability of the fluid, that is, the quantity and quality; (2) the design modifications or design compromises of the source and possible increased complexity; (3) the overall reliability; (4) the design requirements for the expendable cooling equipment; and (5) the advantageous over other means or approaches to cooling.

In general, the utilization of generated fluid should result in weight savings or in design simplicity or the combination of the two over that of other possible means for achieving the desired temperature control.

Residuals - The use of residual fluids and boil-off from cryogenic tanks as expendable coolants may be practical if sufficient heat sink capacity (heat of vaporization + temperature rise x capacitance) is available. Also, it will depend on the weight of a residual fluid cooler system to other systems.

Various physical characteristics such as heat of vaporization and an estimate of cooler weight (compared to equivalent water boiler) of residuals is given in Table 9.

An examination of the cooler weight indicates that the weight (and size) do not change greatly with the residual fluid. This is true because the circulating electronic coolant governs much of the design by available coefficient and wall temperature requirements. These estimates were based on a 60% glycol-40% water solution and could change if a coolant with different properties and freezing temperature were used. The closeness to existing water boiler sizes permits the selection of the most suitable type cooler for each residual.

Fluids with freezing temperature and vapor pressure close to that of water probably would be used in a cooler similar to a sublimator. H_2O_2, N_2H_4 and possibly N_2O_4 or 50-50 N_2H_4-UDMH fall into this group. Fluids with lower freezing temperature, with low vapor pressure and/or heat of vaporization, would probably require a plate or wick type boiler. MMH, B_5H_9, HNO_3 etc. fall into this group. Some improvement

Table 9. Physical Properties of Residuals

	h_v	t_F	t_B	p_B	t_c	p_c	sg	c_p	k	μ	W_c/W_{cH_2O}
Fuels											
H_2	195	-434	-423	NA	-400	188	NA				.7
NH_3	596	-107	- 28	73.3	270	1636	.64	1.10	.200	11	.7
N_2H_4	540	35	236	.7	716	2131	1.02	.73	.21	82	1.25
MMH	377	- 63	189	.3	609	1195	.88	.69	.146	85	1.4
B_5H_9	219	- 54	140	1.6	435	557	.64	.53	.10	25	1.3
50% N_2H_4 / 50% UDMH	426	18	170	.65	635	1696	.91	.68	.152	82	1.2
Oxidizers											
H_2O_2	596	31	302	.077	855	3146	1.45	.63	.28	100+	1.6
HNO_3	216	- 43	181	1.3	549	1240	1.55	.423	.20	73	1.4
ClF_3	128	-105	53	11.0	345	838	1.87	.300	.137	35	1.2
N_2O_4	178	12	70	7.0	316	1470	1.48	.352	.08	34	1.2
ClF_5	73	-153	8.4		289	771	1.9	.32	.13	1.3	1.2
NF_3	73	-341	-201	NA	- 88	729	NA				.7
O_2	92	-362	-297	NA	-182	731	NA				.7
F_2	72	-365	-307	NA	-201	808	NA				.7
85% HNO_3 / 15% NO_2	247	- 56	148	.9	520	1286	1.55	.417	.172	120	1.3
Pressurants											
N_2	85.8	-345	-320	NA	-232	492	NA				.7
He	10.5	-456	-452	NA	-450	32	NA				.7

h_v = heat of vaporization, Btu /lb
t_F = freezing temperature, F
t_B = boiling temperature, F at 14.7 lb/sq in
p_B = saturation pressure at 40 F, psia
t_c = critical temperature, F
p_c = critical pressure, psia
sg = specific gravity of liquid 40 F
c_p = specific heat of liquid at 40 B, Btu/(lb)(F)
k = thermal conductivity of liquid, Btu/(hr)(sq ft)(F)
μ = viscosity, (lb)(sec)/(sq ft)
W_c/W_{cH_2O} = estimate of weight of the cooler using this residual divided by the weight of a wick type water boiler cooling circulating fluid to 50 F at the same heat rejection

in boiling coefficient occurs as the vapor pressure rises. Plate type coolers or possibly a tubular forced vortex cooler would prove superior here. ClF_3 and ClF_5 belong in this group. When both the vapor pressure and boiling coefficients are high such as (NH_3), a tubular forces vortex cooler would be the most practical. The cryogenic fluids would be utilized in a gas-to-liquid heat exchanger, such as the concentric tube type.

An excellent example of the utilization of residual and boil-off is given in the discussion in the next section on the utilization of hydrogen vent gas.

Design Concepts

Figure 27 illustrates three general concepts which utilizes solid or liquid expendable coolants, and the general usefulness for each are given. The selection of the particular concept will depend mainly upon the cooling load and duration. The subliming solid concept has been included, but it may not be feasible for the mission durations considered in this study. The two concepts for liquid expendables indicate the possibility of utilizing either stored liquid or residual or the combination of the two.

Various designs for the heat exchanger for the expendable liquid coolant are illustrated in Figure 28. All utilize standard heat transfer surfaces. Evaporating water-glycol heat exchangers have been studied for a number of years by several different organizations. A considerable backlog of information has been accumulated. However, very little information on boiling with water containing desolved solids is available. Both the sublimator and wick type boilers, which have received themost attention, have small passageways which could be plugged by the solids in the water. Further work on evaporating water cooling is being undertaken under Contract NAS 8-11291 by AiResearch Manufacturing Co. for NASA-Huntsville.

For design purposes, the following approximate weight equation for a water-glycol evaporative boiler may be used:

$$W_{WB} = .0006 + \frac{2}{h_v \rho_{v@T_s} P_c}$$

where W_{WB} = weight in lbs per Btu/hour heat dissipation

and $\rho_{v@T_s}$ = density, lb/cu ft of the saturated evaporant vapor at the glycol temperature

h_v = heat of vaporization of fluid, Btu/lb

P_c = critical pressure of fluid, lb/sq in

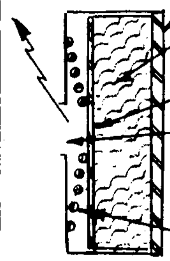
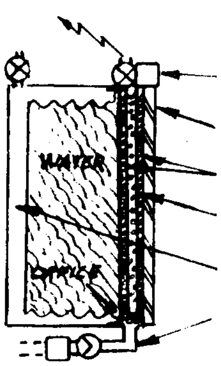
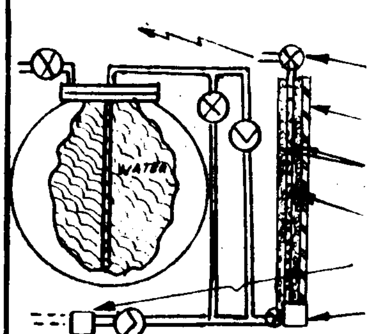
Figures	Region of Usefulness
 Subliming Solid Concept	Moderate to low heat dissipation Short time Rough temperature control Restricted temperature range Premium for extreme simplicity
 Contained Expendable Concept	Moderate to high heat dissipation Fine temperature control Extended temperature range Testability required
 Separate Tank Expendable Concept	Moderate to high heat dissipation Testability required Minimum coldplate package volume

Figure 27. Expendable Cooling Concepts

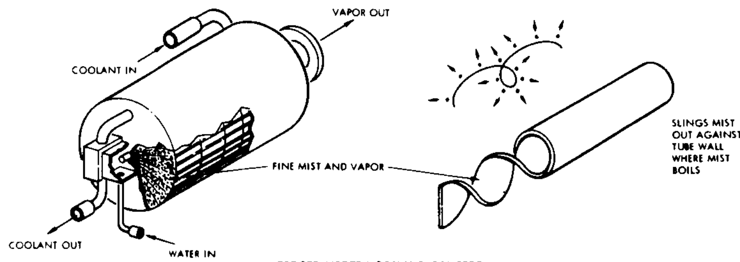
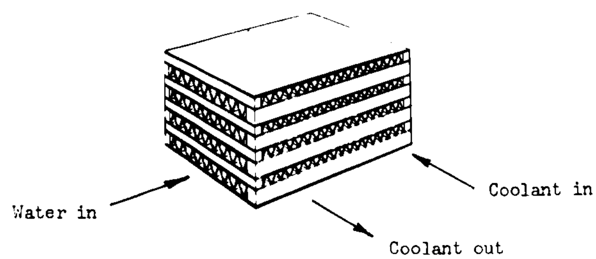
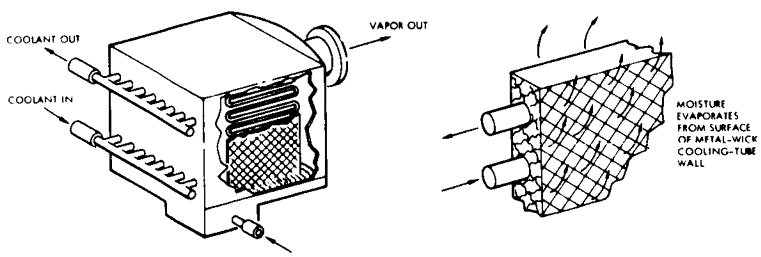
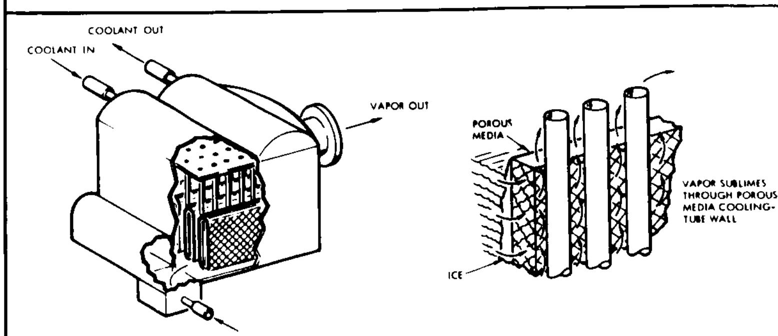
Figures	Region of Usefulness
 FORCED VORTEX BOILING CONCEPT	High boiling coefficients h_B min > 100 h_B avg > 300 Coolant above 125°F using water Coolant above 0°F using ammonia
 PLATE FIN BOILING CONCEPT (Alternate core for forced vortex boiling concept)	Moderate boiling coefficients h_B min > 30 h_B avg > 100 Coolant above 60°F
 WICK BOILING CONCEPT	Same as Plate Fin Boiling Concept
 SUBLIMING CONCEPT	Low boiling coefficients h_B min > 50 h_B avg > 150 Coolant below 70°F using water

Figure 28. Expendable Coolant Equipment - Active Systems

Control Concepts

The controls can be fairly simple but as greater efficiences are required, they tend to become heavier and more complex. A control on the fluid temperature into the electronic equipment can be achieved by controlling the coolant temperature out of the cooler or heat sink. Any fluctuations will be largely absorbed by the capacitance of the system between the cooler and the electronic equipment. The controls of the fluid-out temperature is accomplished in different ways for different coolers. These are described briefly in the following paragraphs.

For the wick type water boiler, Figure 29, the fluid temperature can be controlled by regulating the back pressure or the wetness of the wick. The large capacitance of contained evaporant generally makes it more desirable to control by means of back pressure.

The sublimator, Figure 30, may be sufficiently self-regulating to require no controls other than a shutoff valve. It may however be necessary to make a circulating fluid bypass around the sublimator with a modulating control valve.

For the plate and vortex type boilers, Figures 31 and 32, the control of the evaporant flow on the basis of the circulating fluid exit temperature with a modulating control valve.

Modulating control valves are inherently stable if the time response of the valve is appreciably greater than the time response of the system being controlled. A fluid bypass has a response time of only a few seconds unless it is extremely large. A plate or vortex type boiler also has a response time of only a few seconds unless it is extremely large. The usual self-powered "Vernatherm" modulating valves have response times from ten to thirty seconds. Simple regulation of the sublimator, plate and vortex boilers is thus possible.

The response time of a wick type water boiler is considerably greater than a bypass or evaporant flow controlled boiler. A back pressure control for a wick boiler is much faster in response time than other means of controlling the circulating fluid temperature. A "slow" modulating back pressure valve can be stable if the wick type boiler is slightly over-designed. The more usual electronic control is shown in Figure 29.

Cryogenic expendable cooling can operate on a demand system as shown in Figure 31. Since the time response of the cooler is very rapid, a self-powered "Vernatherm" modulating type control element can be used to give a simple reliabile control.

The various control concepts discussed above operate on a demand basis. The flow of the expendable coolant is regulated to maintain the recirculating fluid exit temperature within the desired range.

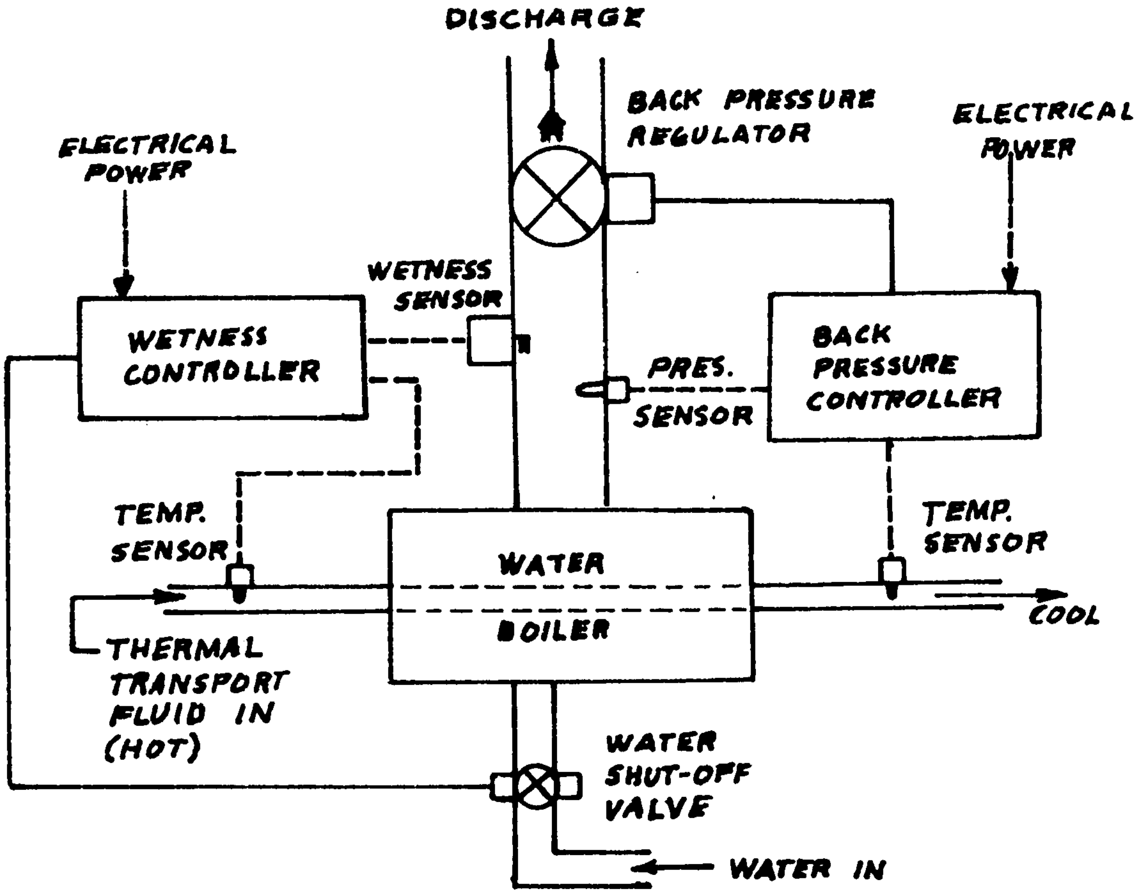


Figure 29. Control Concept for Wick Water Boiler

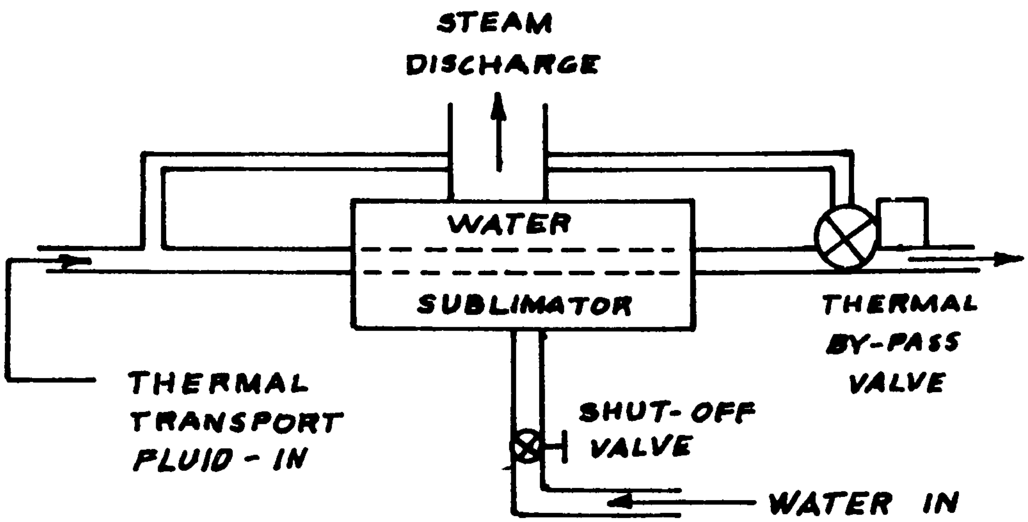


Figure 30. Control Concept for Water Sublimator

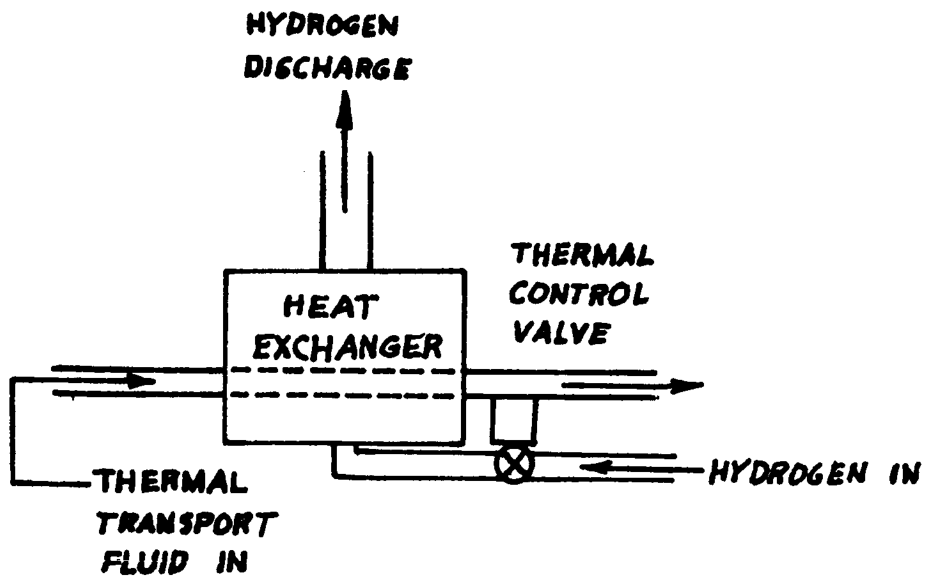


Figure 31. Control Comcept for Cryogenic Gas Heat Exchanger

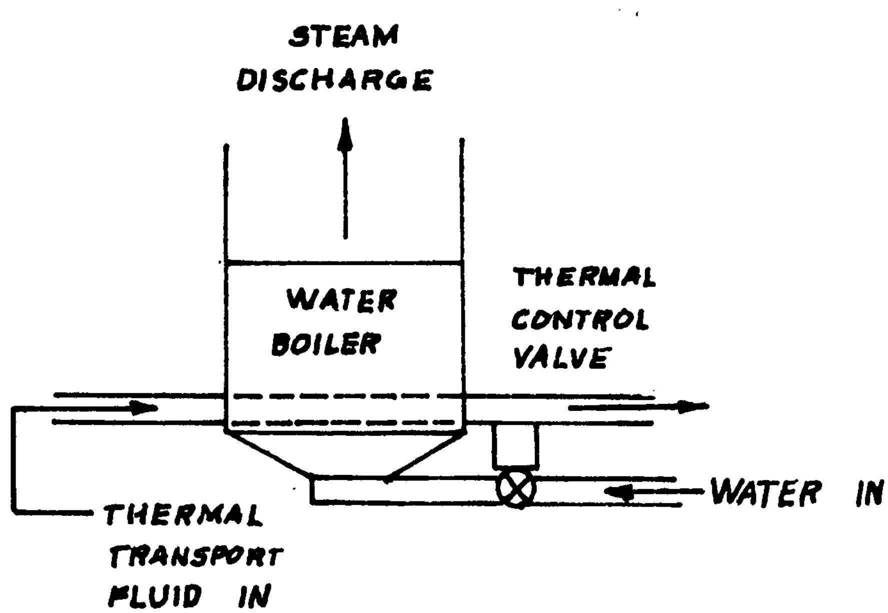


Figure 32. Control Concept for Plate or Vortex Boiler

2.3 PARAMETRIC STUDY AND TRADE-OFF ANALYSIS

Parametric study was initiated in several areas which are basic to a thermal control system. The parametric effort is directed toward providing data in easy to use form which can be readily used for either performance evaluation or comparative evaluation in the selection and sizing of the various components which can then be analyzed more in detail.

Those areas in which some significant results have been obtained are discussed in the following paragraphs. The parametric study is a continuing effort and it is expected that many areas will be investigated.

2.3.1 Application of Thermoelectric Cooling and Temperature Control

The previous quarterly report (Reference 1) stated that thermoelectric (T/E) refrigeration was an excellent method for obtaining heat transfer and temperature regulations within astrionic equipment packages. (Also see paragraph on Reversible Heating/Cooling Control Systems). Once it is determined that T/E cooling is within the scope of the design requirement, it is necessary to determine the T/E requirements such as the number and type of modules and the current and operating power to satisfy these design requirements. Both experimental and analytical investigations have been conducted concerning thermoelectric heating and cooling. These studies have resulted in the "Experimental Parameter Determination Method" which outlines a parametric procedure for determining the performance of T/E devices in conjunction with integral coldplates and heat exchangers of the equipment package. The Experimental Parameter Determination Method is discussed in the following paragraphs.

(For definition and background data on T/E, see equipment cooling reference books and design guides such as NavWep 16-1-532, "Methods of Cooling Electronic Equipment".)

Experimental Parameter Determination Method

(a) The Seebeck coefficient, α, the electrical resistance, R, and the thermal conductance, K, of various commercially available modules are determined by separate, simple experimental tests within the temperature range of interest and expressed as functions of the average temperature. Hence, obtain:

$\alpha = (\bar{T})$ Seebeck Parameter

$R = R\ (\bar{T})$ Electrical Resistance Parameter

$K = K\ (\bar{T})$ Thermal Conductance Parameter

(b) Starting with the basic thermoelectric equation

$$Q_{net} = \alpha I T_c - K \Delta T - \frac{1}{2} I^2 R \qquad (1)$$

where:

Q_{net} = Heat Pumped Per Module

I = Supply Current

T_c = Cold Side Temperature

$\Delta T = T_h - T_c$

(T_h = Hot Side Temperature),

and using the relationship

$$T_s = T_h - Q_{Total}\, R_f \qquad (2)$$

where:

T_s = Coolant Supply Temperature

R_f = Heat Dissipator Equivalent Thermal Resistance (includes thermal contact resistance on hot side)

$Q_{Total} = (EI + Q_{net})\, N$

where: N = Number of Modules

EI = Electrical Input Power

E = Voltage Across One Module,

express Equation (1) explicit for T_h as:

$$T_h = \frac{I \alpha T_c - \frac{1}{2} I^2 R - Q_{net}}{K} + T_c \qquad (3)$$

(c) Substituting Equation (3) into Equation (2) to obtain:

$$T_s = \frac{I\alpha T_c - \frac{1}{2}I^2R - Q_{net}}{K} - R_f\ N(EI + Q_{net}) + T_c \qquad (4)$$

(d) Next, substitute the parametric relationships for α, K, and R into Equation (4) to obtain:

$$T_s = \frac{\alpha(\bar{T})\ T_c I - \frac{1}{2}I^2R\ (\bar{T}) - Q_{net}}{K\ (\bar{T})} - N\ R_f \left[Q_{net} + I^2R\ (\bar{T}) + \alpha(\bar{T})\ I\Delta T\right] + T_c \qquad (5)$$

(e) The iterative solution of Equation (5) produces T_s versus I plots for various values of R_f. R_f as used above is related to the coldplate thermal design as follows:

$$R_f = \frac{1}{\epsilon W\ C_p} = \frac{T_h - T_s}{Q_{Total}}$$

where:

$$\epsilon = \text{Dissipator Effectiveness} = \frac{(T_{out} - T_s)}{(T_h - T_s)}$$

C_p = Coolant Specific Heat

W = Coolant Mass Flow Rate

T_{out} = Coolant Exit Temperature

(f) The T/E input voltage per module is given by:

$$E = \alpha(\bar{T})\Delta T + I\ R\ (\bar{T})$$

and the T/E input power per module is EI. A typical T/E input power versus coolant supply temperature was given as Figure 13 in previous quarterly report, Reference 1.

A typical block diagram of a thermoelectric temperature control system is shown in Figure 33 for a stable platform (IMU) example. The key element in this control system is the thermoelectric heat exchanger, which may be considered to consist of two high-fin density coldplates mounted on both sides of a set of thermoelectric elements. Heat is transferred from the internal recirculated gas stream to the internal coldplate by forced convection where it is thermoelectrically pumped to the hotter external coldplate which in turn rejects heat to the coolant supplied by the thermal control system. By using this approach, the supplied coolant can be of a higher temperature than that of the internal gas.

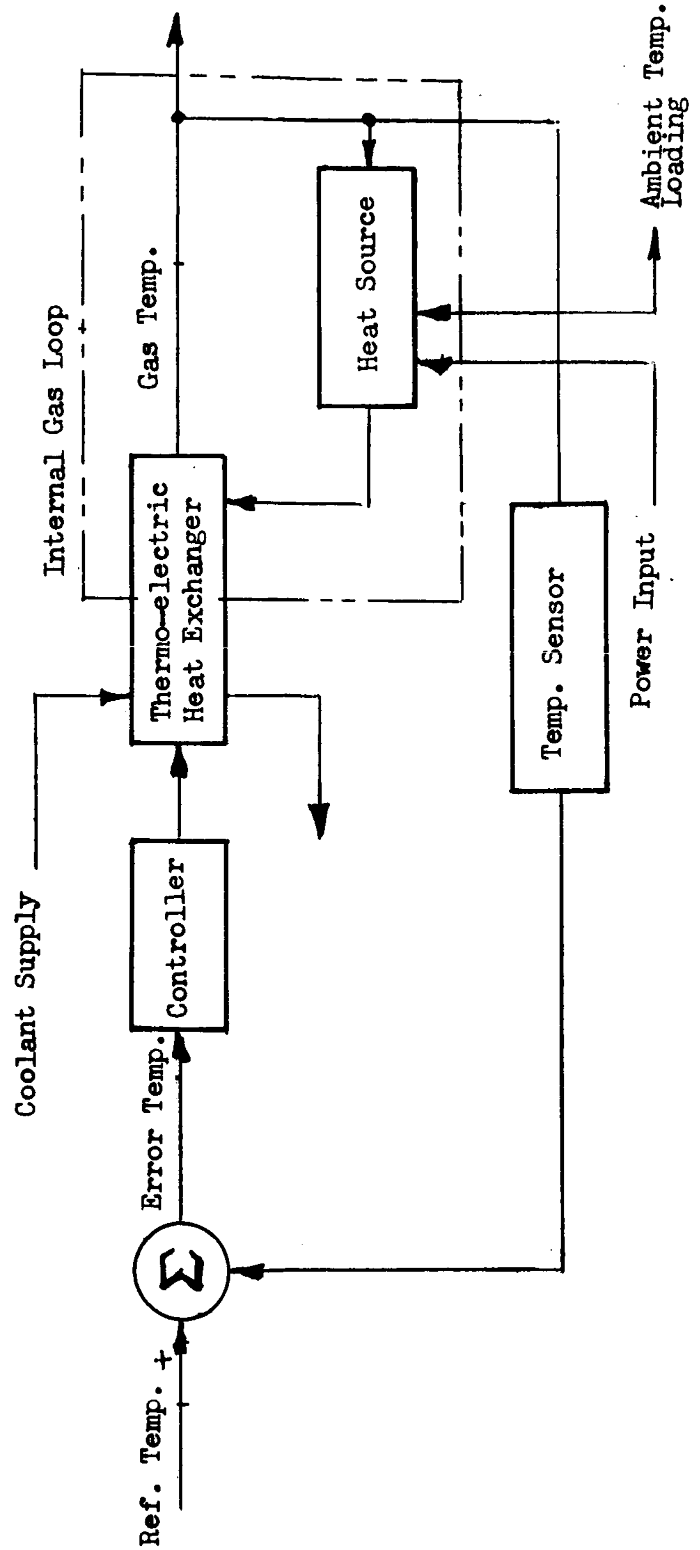


Figure 33. Diagram of a Typical Thermoelectric Control System

2.3.2 Integral Coldplate Concept Versus Vehicle Coldplate Concept

In the last quarterly report, it was postulated that the integral coldplate concept, as distinguished from the vehicle coldplate concept, would become increasingly dominant. This has been substantiated in the industry survey conducted thus far, at least from the astrionic equipment manufacturer's viewpoint.

Tradeoff Considerations

The primary advantages of the vehicle fixed coldplate concept are: (a) coolant lines would not be disconnected in the event of package maintenance or replacement, thereby permitting unreplaced packages to remain operational, and (b) a standardized coldplate design is achieved allowing minimum development and lower unit cost. These advantages must be traded-off against the increase in weight and size due to this approach. Weight and size increases are obvious because at the vehicle coldplate - package interface, there are requirements for two flat, rigid, heavy walls which are replaced in the integral coldplate concept by one wall which need not be flat nor rigid. Also, a standardized concept is always a compromise in which the design is for the limiting worst-case and is over-designed (over-weight, etc.) for the general case. It has been found in numerous instances, the entire weight of the vehicle coldplates can be saved by changing to the integral coldplate concept.

The integral coldplate concept may be required also from another aspect. In future microminiaturized equipment with increased heat densities, there will probably be the requirement of bringing the coolant closer, physically and thermally, to the heat source. For example, in very fast future computers of several nanosecond speed, all the active devices of the computer will have to be only inches away from each other. Such forced microminiaturization, with resultant high-heat density, will require that the heat sink (coolant) be very close to these active devices, in the order of 1/4 inch for conduction to be utilized. Such a design could be achieved by distributing a number of small integral coldplates within the package. The integral coldplate and heat exchanger concepts provide an additional advantage of serving as a primary or secondary package structures since they have high strength-to-weight ratio design.

Also, the replaceability of astrionic packages can be retained in the integral coldplate concept by: (a) use of self-sealing quick disconnects, or (b) by use of plug-in modules which can be replaced. Self-sealing disconnects which are small and reliable and which have low flow-pressure loss and leakage characteristics are commercially available. Many are presently used on space vehicles. Plug-in modules are commonplace today and will be used with greater frequency.

This is because fault isolation to the module level is increasingly possible due to microminiaturization. Even without fault-isolation to the module level, it is possible to remove and replace all the plug-in modules of a package which has failed, leaving in place the low failure rate package chassis and its integral coldplate.

Interface Control Methods

Figure 34 depicts two different methods of specifying (controlling) the thermal interface between the astrionic packages and the vehicle. The two methods, the integral coldplate concept and the vehicle coldplate concept, are related to forced convection (Column II of Figure 19) and conduction (Column III of Figure 19) modes of package external heat transfer, respectively.

Parameters which are specified in the integral coldplate concept include:

(a) Required coolant flow versus coolant supply temperature curve.

(b) Coolant pressure lead versus coolant flow curve.

(c) Ambient temperature and absolute pressure ranges.

(d) Mechanical mounting location and increases.

(e) Hydraulic convection, e.g., type of quick disconnects.

Typical curves for (a) and (b) above are illustrated by Figures 35 and 36.

Parameters which are specified in the vehicle coldplate concept include:

(a) Coldplate temperature or temperature distribution.

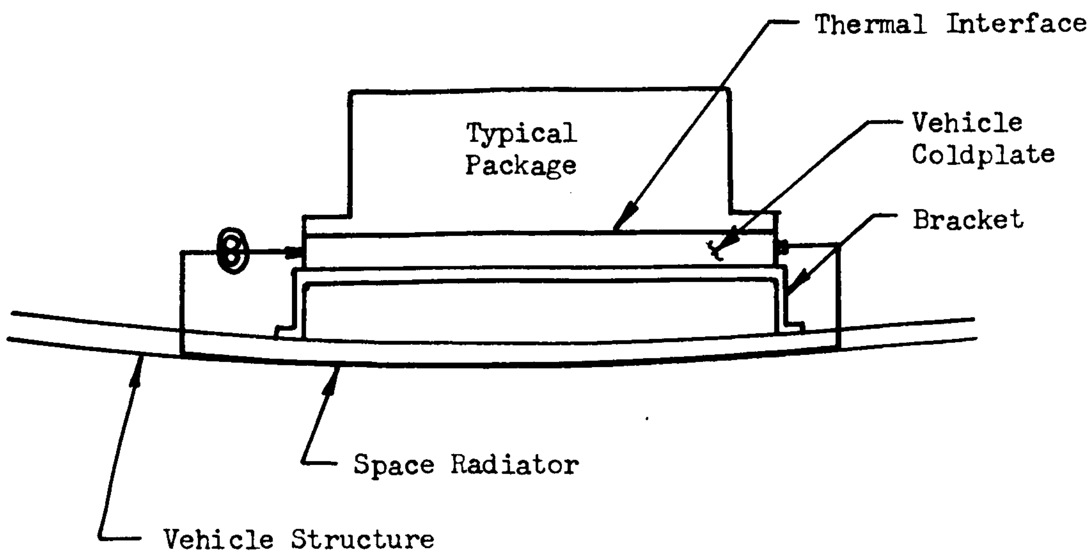
(b) Average and maximum heat flux.

(c) Mechanical mounting location and means.

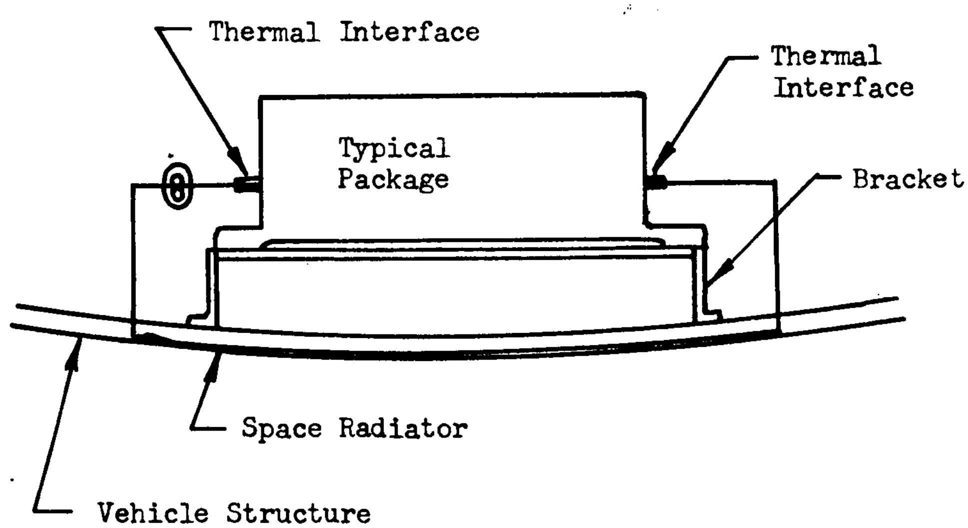
(d) Special mounting features such as available area, flatness, finish, and requirement for interface material (greases).

(e) Ambient temperature and absolute pressure ranges.

(f) Mounting procedures such as bolt torque values and sequence of torquing.



(a) Vehicle Coldplate



(b) Integral Coldplate

Figure 34. Electronic Package Thermal Interface Methods

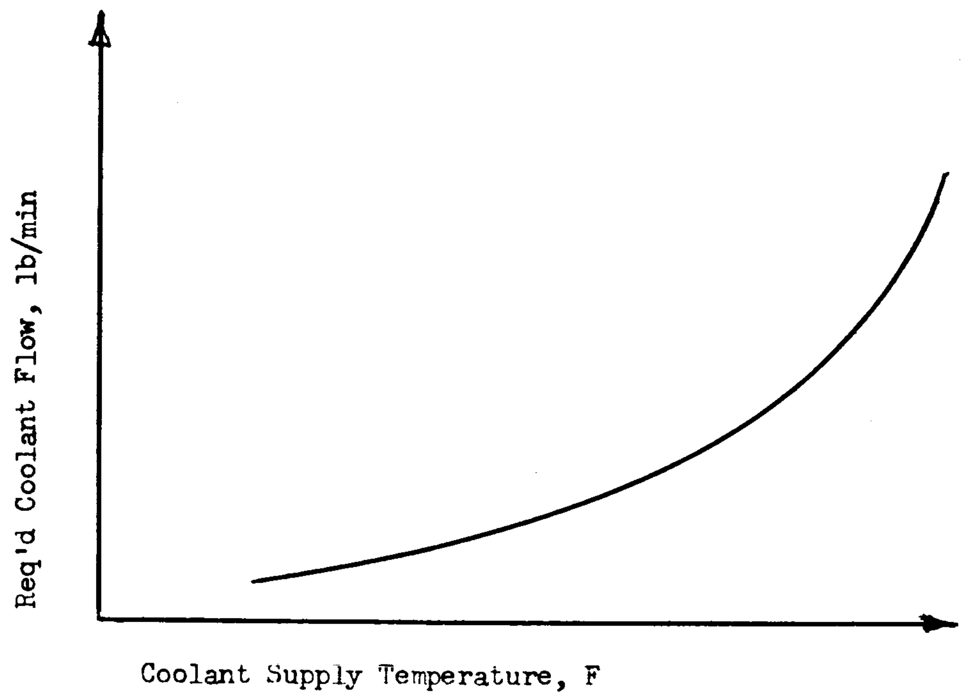


Figure 35. Typical Coolant Flow Versus Supply Temperature Interface Requirement

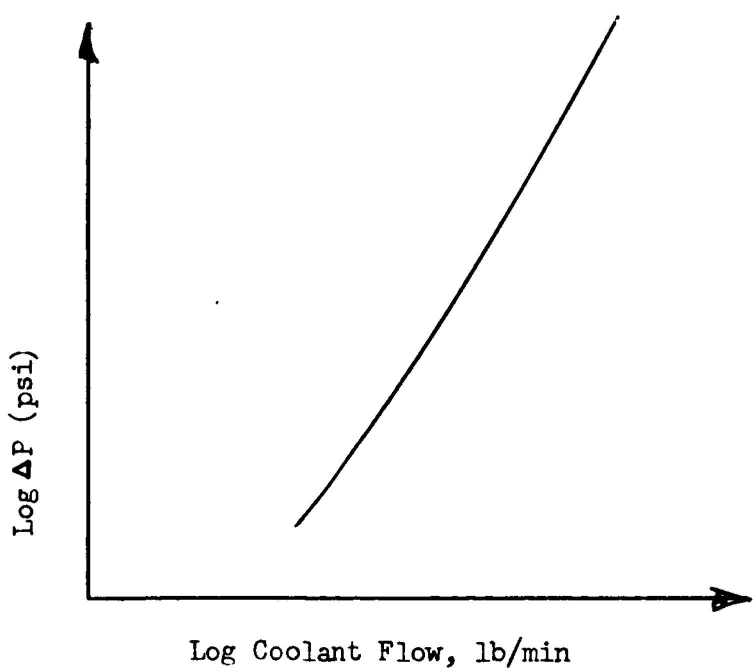


Figure 36. Typical Coolant Flow Versus Pressure Drop Interface Requirement

2.3.3 Utilization of Hydrogen Boil-off

The possible proximity of the astrionic equipment to a Saturn upper stage, such as the S-IVB or S-II, suggests the possible use of residual cryogenic boil-off for equipment cooling. The example of hydrogen boil-off utilization is discussed in the following paragraphs.

For a booster stage in space which contains residual hydrogen, heat transferred to liquid hydrogen from solar and other sources causes the hydrogen to boil-off and increase in temperature and the tank pressure. This process will continue until the tank design pressure is reached. At this point, the pressure is controlled by venting hydrogen gas through the tank pressurization valve. This is illustrated by curve A of Figure 37. The vent gas flows from the tank at approximately saturation pressure (design tank pressure) and temperature (always less than -400 F). For an estimated initial liquid enthalpy of about 120 Btu/lb, the enthalpy of saturated hydrogen vapor varies from about 306 Btu/lb at 0.6 atmospheres (and also at 8 atmosphere) to about 317 Btu/lb at 3 atmospheres. Thus, heat removed from the liquid hydrogen in the tank by evaporation varies little with tank pressure. The T-S hydrogen diagram, Figure 38 shows this. This is also shown in Figure 39 for the case of H_i = 108, where H_i is the initial enthalpy of liquid hydrogen.

Based on the above estimates for saturated hydrogen gas, approximately 1200 Btu's may be absorbed by one pound of hydrogen for a temperature rise from saturation conditions to 0 F. This provides a very effective heat sink for equipment cooling.

Prior to reaching the tank design pressure, the hydrogen boil-off may be vented in different ways, as indicated in Figure 37. Curve A of Figure 37 represents the case of no venting until the design pressure is reached and thus limits the utilization period. The other three approaches indicate how the boil-off may be utilized from the point of initial pressure build-up. There may be other uses for the hydrogen boil-off besides equipment cooling which will influence the approach to be used.

The desired approach for hydrogen utilization is the demand basis and this is indicated in the following analysis. For a heat load profile such as Figure 13, a hydrogen-fluid cooler designed to handle a uniform base load of 100 watts would reduce the heat load for a space radiator from 2000 watts to 1900 watts, which requires a radiator area of 130 square feet. If this same amount of hydrogen for cooling was available on a demand basis, half of the peak or spike loads could be handled by the hydrogen cooler and this would reduce the radiator (cylindrical normal to sun at one point) requirements to about 750 watts. This would mean a radiator area of 51 square feet, which is a sizeable reduction in radiator area. If there is sufficient hydrogen

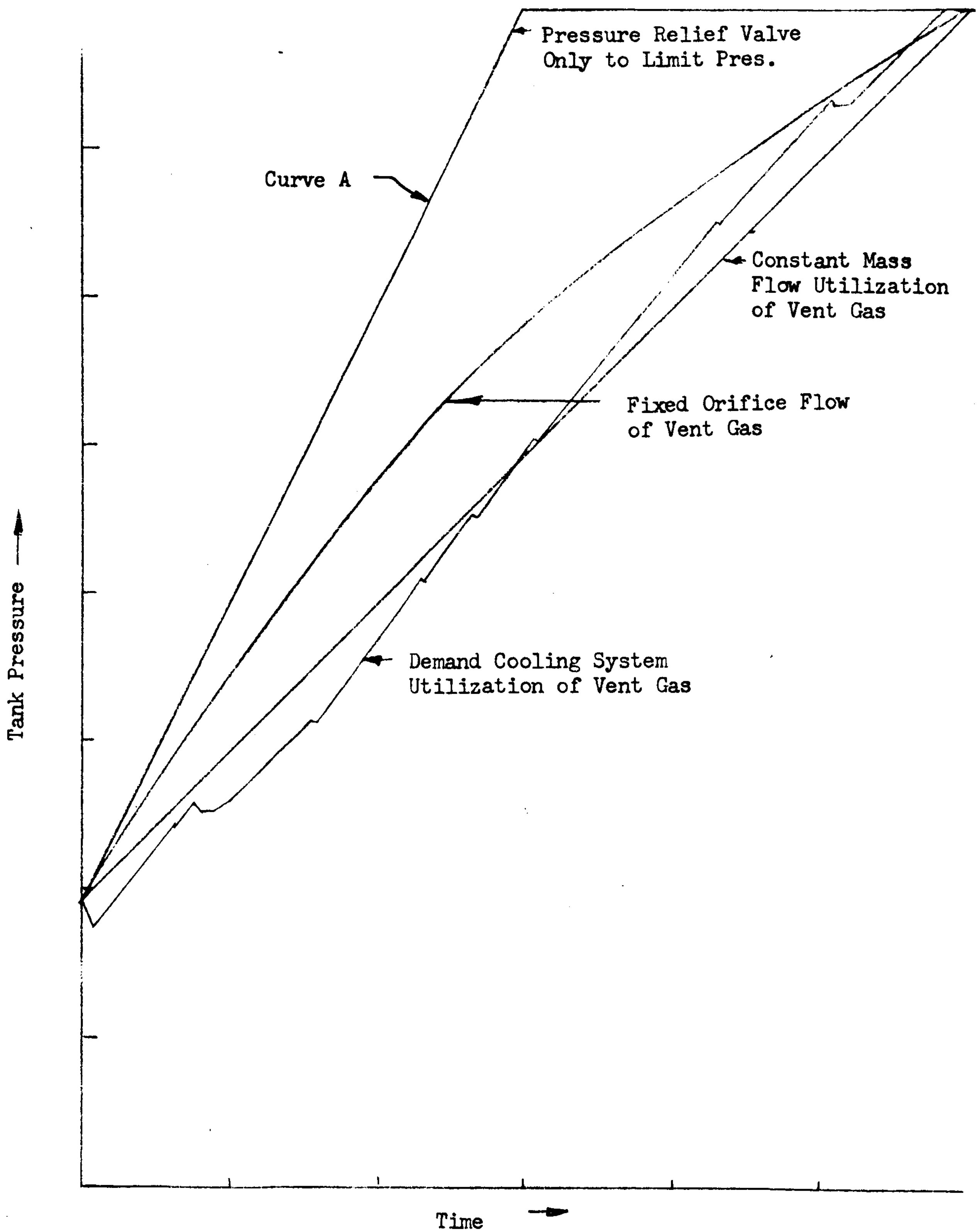


Figure 37. Tank Pressure Versus Time

Reference: NBS Heat & Power Div. Research Paper 1432

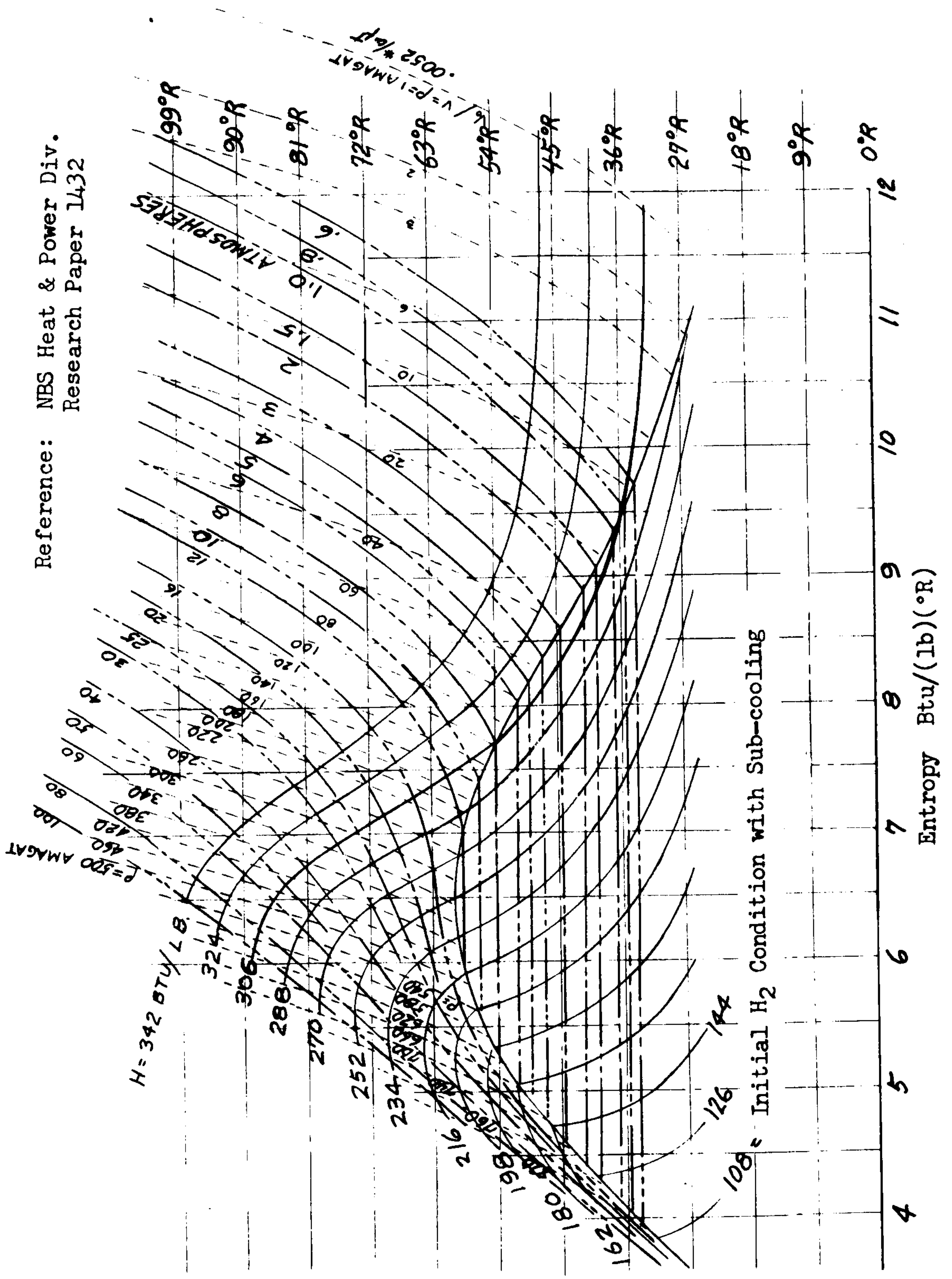


Figure 38. Hydrogen T-S Plot

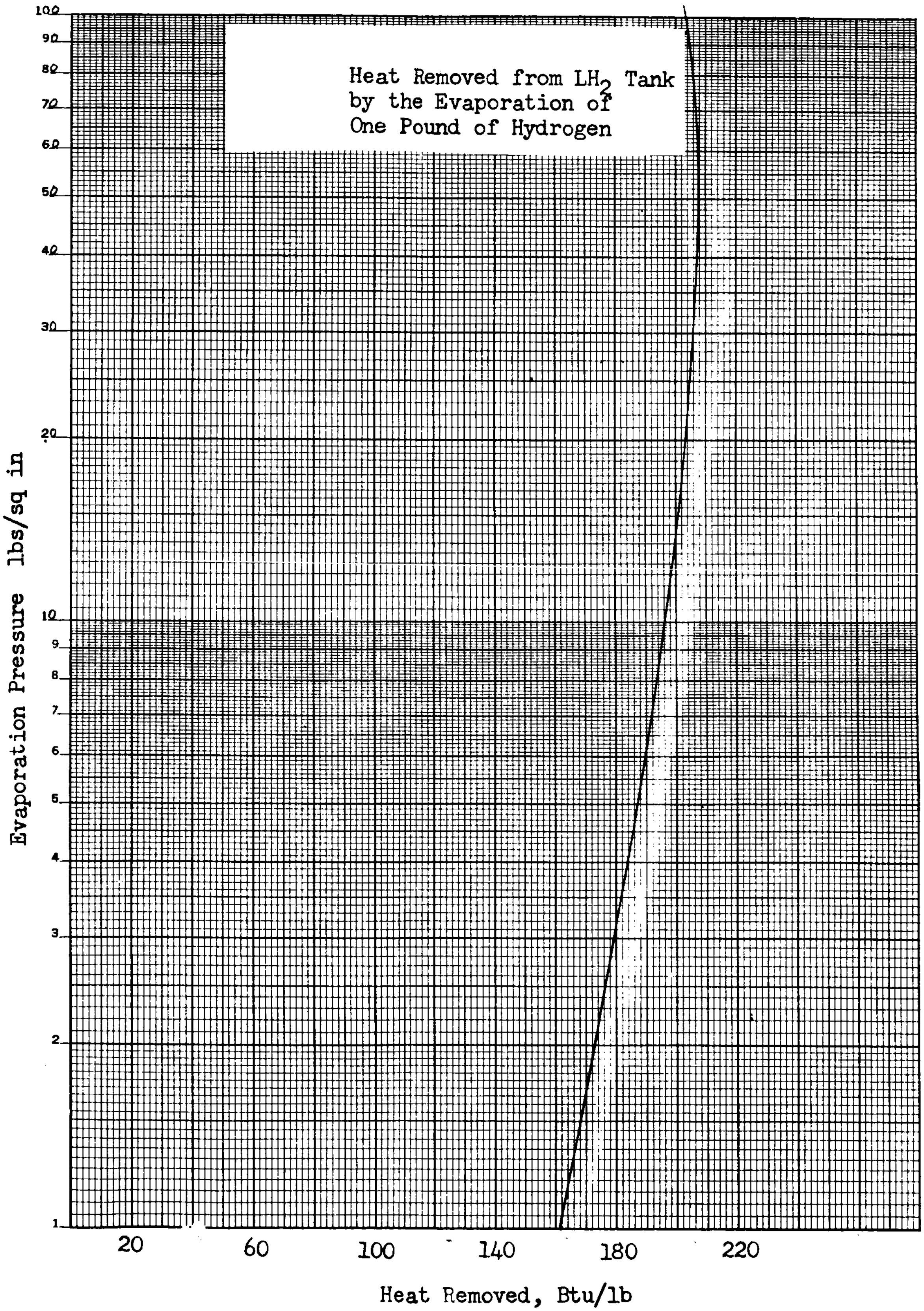


Figure 39. Hydrogen Evaporation Pressure Versus Heat Removed

to handle a uniform base load of 200 watts, the radiator heat load would be reduced from 2000 watts to 1800 watts, which requires a radiator area of 124 square feet. If this same amount of hydrogen for cooling was available on a demand basis, the hydrogen cooler would be able to handle all the peak or spike loads and would reduce the radiator heat load to 300 watts. The radiator area would be 20.4 square feet. This clearly indicates the desirability of a demand basis for hydrogen utilization.

All systems utilizing hydrogen vent gas on a demand basis to take up part or all of the cooling load of the electronic equipment can have system advantages over any type of cooling system. For example, the use of the vent gas continuously (though non-uniformally) would reduce the necessary size of the pressure relief valve and vent lines when they are sized by conditions in space. Another example, the possible demand nature of the system may allow a considerable reduction in radiator area or elimination of radiators since the peak loads are taken care of by the hydrogen-glycol cooler. The hydrogen could also supply the pumping power for a liquid coolant cirucit.

A cursory examination may show that the total venting is more than sufficient to handle cooling requirements. It may however be necessary to carefully determine total energy dissipation by the electronic equipment. If this energy can be dissipated by the allowable boil-off, only a hydrogen-to-circulating fluid heat exchanger is necessary. This would be the simplest, lightest and most reliable heat sink for equipment cooling.

In those cases when the hydrogen vent gas cannot meet the total cooling requirement or a portion of the available gas is diverted for other purposes, the available cooling capacity may be used for peak heat load conditions. This is particularly ideal from the standpoint of using radiators to take care of the sustained loads and the hydrogen circulating fluid cooler to take care of peak loads because each is utilized under its most suitable conditions.

If, for longer duration missions, the energy of isentropic expansion of the hydrogen vent gas is also removed from the remaining hydrogen liquid, it will make little difference at what pressure the evaporation takes place (based on a fixed expansion ratio). It is thus apparent that the point in time when the venting takes place will not materially affect the quantity of gas vented. The vent H_2 gas (if used in an expander) will be vented at very low pressure which will severely restrict the use of vent hydrogen gas for cooling. Only low hydrogen pressure drop systems (Figure 40, type 1 and 2) could be used. An efficient expander which has the flow utilized for cooling on a demand basis would have a lengthy developmental period.

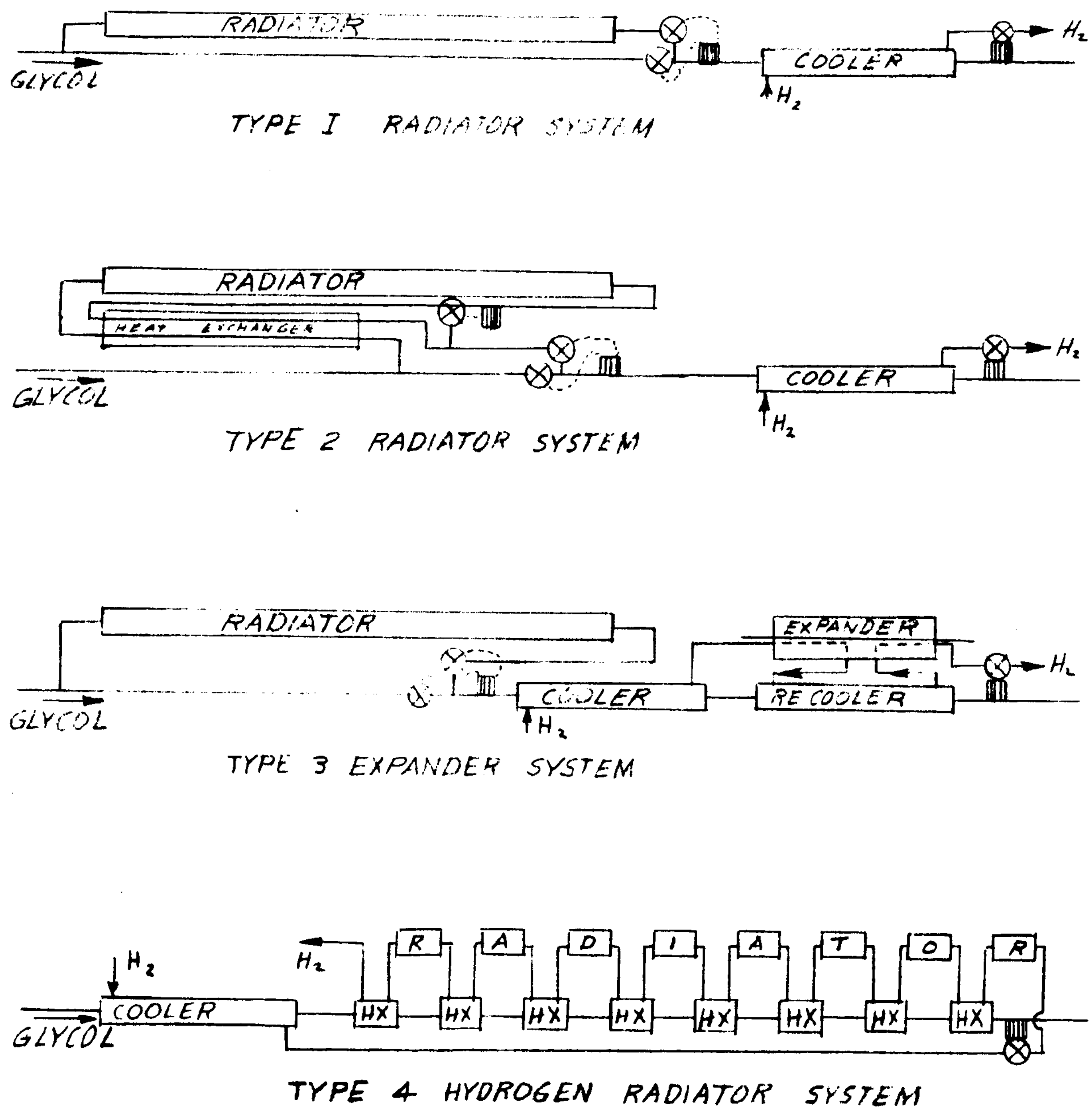
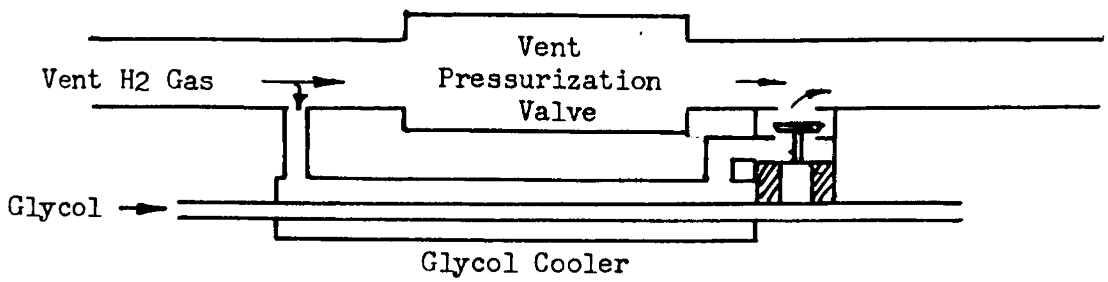


Figure 40. Hydrogen Vent Gas Concepts for Cooling Electronic Equipment

One of the important design consideration for the utilization of the hydrogen vent gas is the location of the hydrogen gas-to-liquid cooler in relation to the vent line and the equipment to be cooled. In general, the cooler should be located in close pro imity to the vent line, to minimize the length of the hydrogen gas duct. It will be simpler to have longer liquid coolant lines from the astrionic equipment to the cooler.

In order to have a controllable flow of hydrogen, the hydrogen must be tapped off upstream of the vent pressurization valve as shown below.



System Concepts

Four possible combinations of hydrogen gas-to-liquid cooler and space radiator are illustrated in Figure 40 and disucssed in the following paragraphs.

Type 1 radiator system is a simple bypass system for the radiator using a highly reliable wax controls element. For example, when the coolant temperature gets below 19 F, the bypass around the radiator opens to control the temperature by adjusting flow thru the radiator. When the coolant temperature gets above 21 F at the cooler exit, the demand portion of the system (also a wax controls element) opens sufficiently to drop the coolant temperature to the desired inlet temperature (20 F). Bypass of the coolant around the radiator cuts the flow in the radiator which drops the radiator exit fluid temperature and reduces the radiator cooling capacity. The minimum heat rejection (without other incident heat inputs) will keep only a certain amount of area above the freezing (or congealing) temperature. As the radiator area increases (when the minimum heat rejection remains constant) the minimum radiator fluid outlet temperature drops. The simple type 1 system thus can be used only for missions of limited number of days unless some means of augmenting the heat supply at min. load is provided or a coolant with lower freezing temperature is used. With mission lengths below some fixed number of days, a radiator may not be necessary.

Type 2 radiator system modifies the Type 1 system to operate the radiator at a more uniform temperature. A heat exchanger, which is bypassed until the radiator outlet temperature approaches the freezing temperature of the coolant, drops the temperature of the coolant into the radiator. Since the outlet temperature is controlled, the inlet temperature can be dropped from possibly 40 F to 0 F or even lower depending on the heat exchanger effectiveness. Since the average radiator temperature is lowered, the heat rejection per unit area is lowered and for a fixed constant heat load, the area of the radiator which will not drop the coolant below some fixed temperature is increased. Type 2 radiator system would only be used at a length of mission greater than Type 1 maximum mission length. This system must compete with a Type 1 system using a coolant which will go to a lower freezing temperature (such as alcohol). The radiator temperature for gaseous heat dissipation versus radiator area is shown in Figure 41.

Type 3 expander system utilizes the hydrogen gas after it has been heated up in the cooler and still is at a reasonable pressure. This gas is then allowed to dowork which will lower the gas temperature. This gas is then passed thru a re-cooler to extract more heat from the circulating fluid. This system can be considered in two areas: - where length of mission slightly exceeds the Type 1 system without a radiator and where a radiator and heat exchanger are needed for the Type 3 system. The expander in this system is in direct competition with the radiator in the Type 1 system. It is doubtful that the expander with moving parts and greater complexity would be sufficiently lighter to favor selection of the Type 3 system. Where the minimum fluid temperature in the radiator drops below the freezing point for longer range missions, the expander reduces the required radiator area and permits operation of the radiator without danger of freezing. The expander competes with a heat exchanger in the Type 2 system for solving the freezing problem. The heat exchanger will probably be lighter as well as less complex. The use of both the heat exchanger and expander would extend the mission length beyond where either alone could. A vortex tube expander though very inefficient does look attractive when combined with a control valve. This cannot be used if an expander is used to reduce H_2 boil-off.

In the Type 4 system, the glycol coolant radiator is replaced by a hydrogen gas radiator. Type 4 hydrogen radiator system requires that the radiator be broken into a number of separate elements, the number depending on the hydrogen flow rate and the amount of radiator area. The temperature difference between the hydrogen and circulating coolant fluid causes the radiator to operate at a lower temperature and thus reject less heat per unit area. This means greater radiator area is required. Since the radiator fluid is an expendable fluid, less meteoroid protection is required which helps to compensate for increased weight due to the greater required area. The radiator has no freeze up problems and the simplest possible controls. It can be used for any mission length and the heat exchangers can very likely replace existing manifolds. The pressure drop on the hydrogen side due to

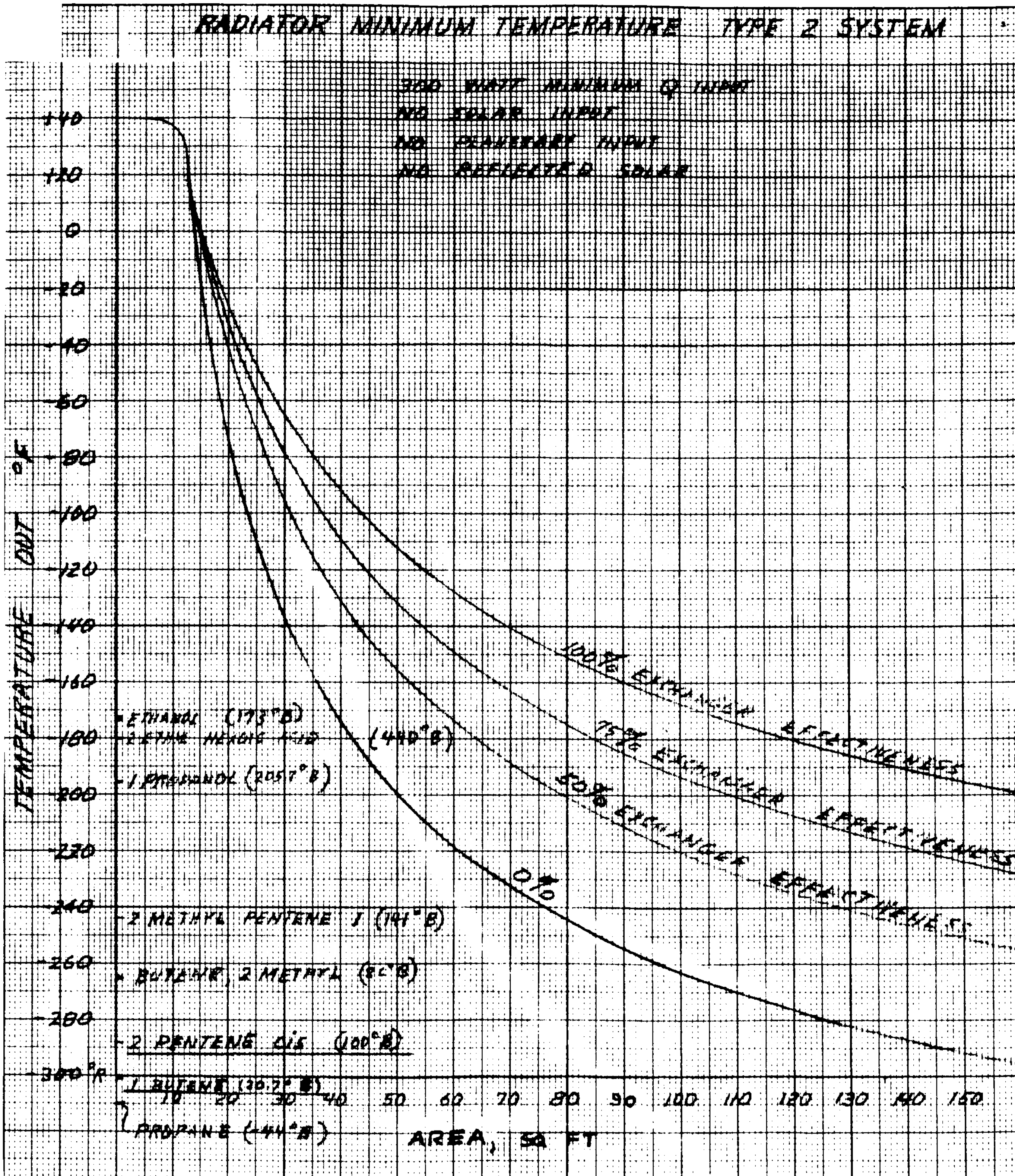
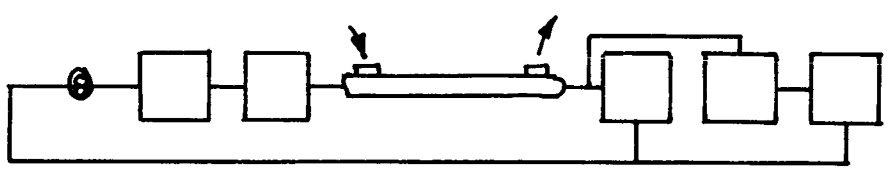


Figure 41. Radiator Minimum Temperature, Type 2 System

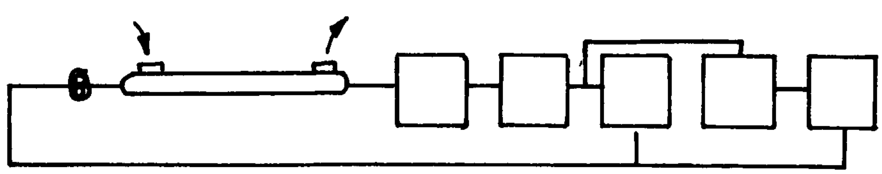
the series arrangement may cause the fluid passages to be considerably larger than equivalent glycol lines. However there is no appreciable wet weight.

If the hydrogen circulating fluid cooler is long in relationship to its diameter, it may be divided into more than one cooler. By this method, it may be possible to locate the cooler where manifolding would be necessary. This would result in a saving of the wet weight of that portion of the manifolds. The vent hydrogen could be ducted to each of the multiple coolers with each controlled separately or to first one then the other cooler. The later, with only one control would be more reliable and have higher coefficients but might also have more weight unless the coolers are close together.

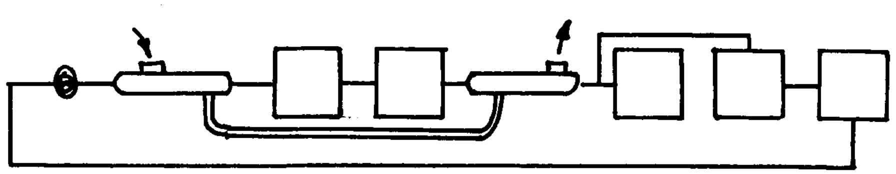
It would be desirable to locate the cooler so it would serve as part of the manifolding. Several possible schematic arrangements are shown below:



Cooler After Coldplate with Maximum Heat Input



Cooler After Pump



Multiple Coolers

Two possible methods for the utilization of hydrogen in a coldplate directly are illustrated in Figure 42. There are numerous problems associated with these methods to make them impractical for further consideration.

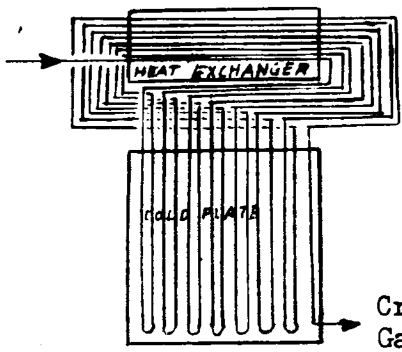
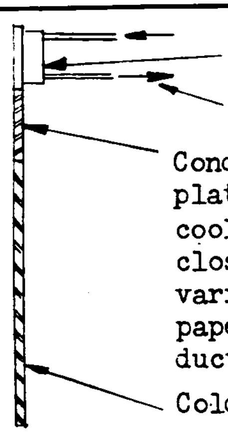
Figures	Description	Region of Usefulness
 Direct Cooled Coldplate Cryogenic Residual Gas	The use of cryogenic gas in a coldplate directly requires that the gas be as close to the desired coldplate temperature as possible. This requires a multi-pass heat exchanger and coldplate as shown. It can be controlled.	Coldplate with big peak loads of short duration Coldplate away from circulating cooling fluid
 Cryogenic Expendable Concept	Controls can be incorporated to regulate flow of gas to control the coldplate temperature	Use of cryogenic residual Premium for extreme simplicity Minimum coldplate package volume

Figure 42. Concepts for Direct Utilization of Hydrogen in Coldplate

Design Analysis

With a load distribution shown in Figure 13 and two different average wattage heat dissipations, it was possible to calculate the hydrogen utilization of a circulating fluid cooler. Figures 43 and 44 show this.

Figures 45, 46 and 47 show how much radiator area is required as a function of mission time and amount of hydrogen boil-off. These figures are based on a permissible or demand utilization of the hydrogen in a cooler with an effectiveness of 0.8 at maximum load.

The hydrogen boil-off permissible for a demand system may be limited by factors other than the heat input to the tank (i.e., total quantity of hydrogen). When utilized in space, the tank may have an available quantity in excess of the estimated boil-off. This could become available by allowing utilization to drop the tank pressure down to the triple point and even to some freezing (slushing) of the hydrogen. If the actual boil-off were treated as the total boil-off the radiator area could be calculated or estimated by interpolation between Figures 45, 46 and 47.

Hydrogen Gas-Glycol Cooler Design

For the proper design of a hydrogen gas-glycol cooler, several important items must be considered. The maximum reliability can be achieved by utilizing a single surface of primary material unbroken by welds, brazing or gaskets between the circulating coolant and space or the circulating coolant and vent hydrogen. Unless an exorbidant penalty is paid, this should be a design criteria. Fins can be brazed or joined to this surface to bring about the desired hydraulic radius and turbulance for either hot or cold fluid. Particular attention to the metal surface temperature can prevent the heat exchanger metal at any point from dropping in temperature below the congealing or freezing point of the circulating fluid. This should be true at off-design conditions and even conditions where the flow rate of the circulating fluid drops off drastically (partial failure in cooling circuit). Attention must also be payed to the thermal expansion since fluids at markedly different temperatures enter the heat exchanger. High effectiveness of the exchanger is required only if the vent hydrogen is in short supply at some time period. Little however could be gained by using effectiveness below 0.8 unless the hydrogen pressure is very low due to the use of slush or subcooled hydrogen liquid.

Based on the above discussion and other considerations, a tentative list of design criteria for a hydrogen gas-glycol cooler has been established:

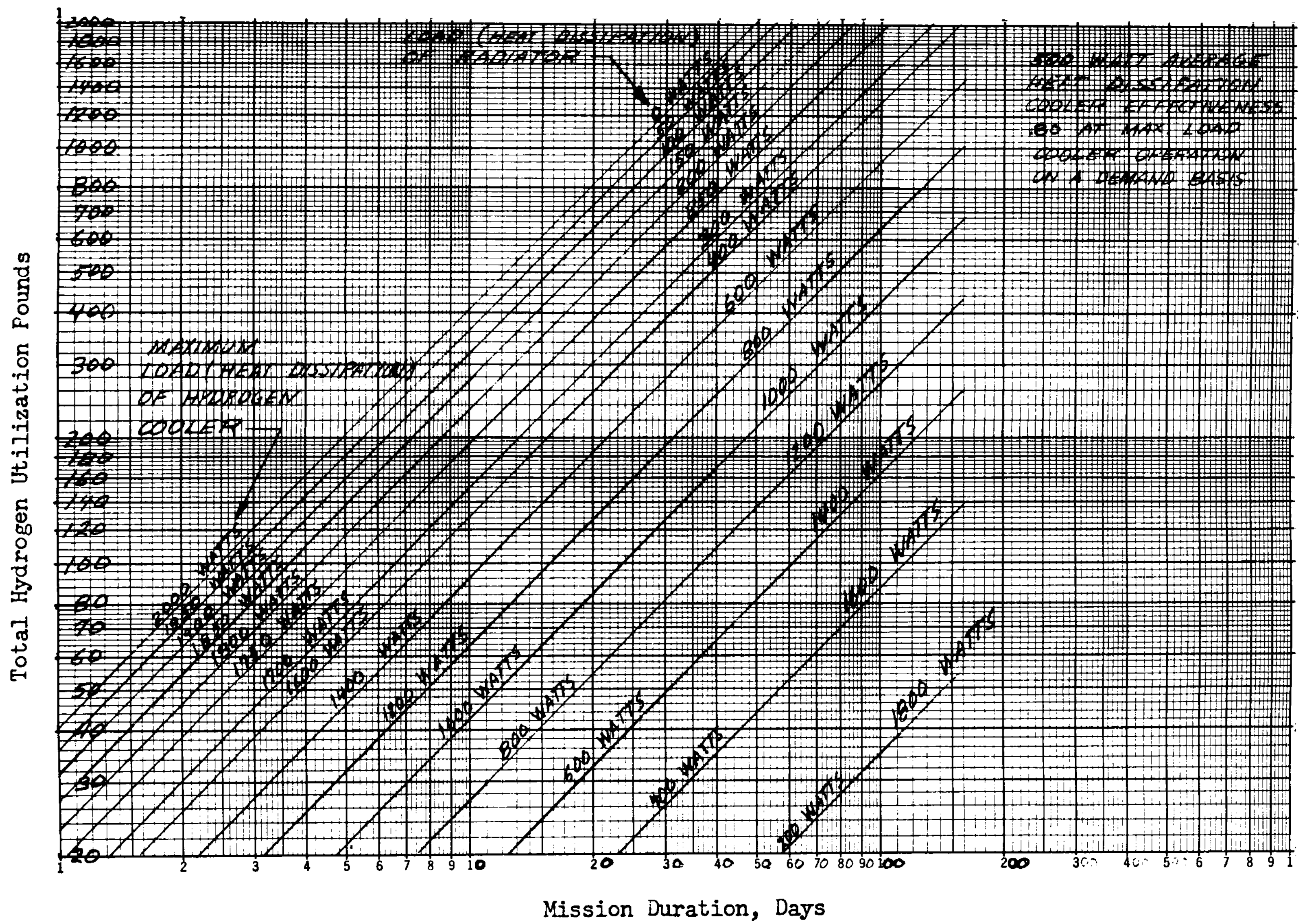


Figure 43. Radiator and Cooler Size as a Function of Hydrogen Utilization, 500 Watt Load

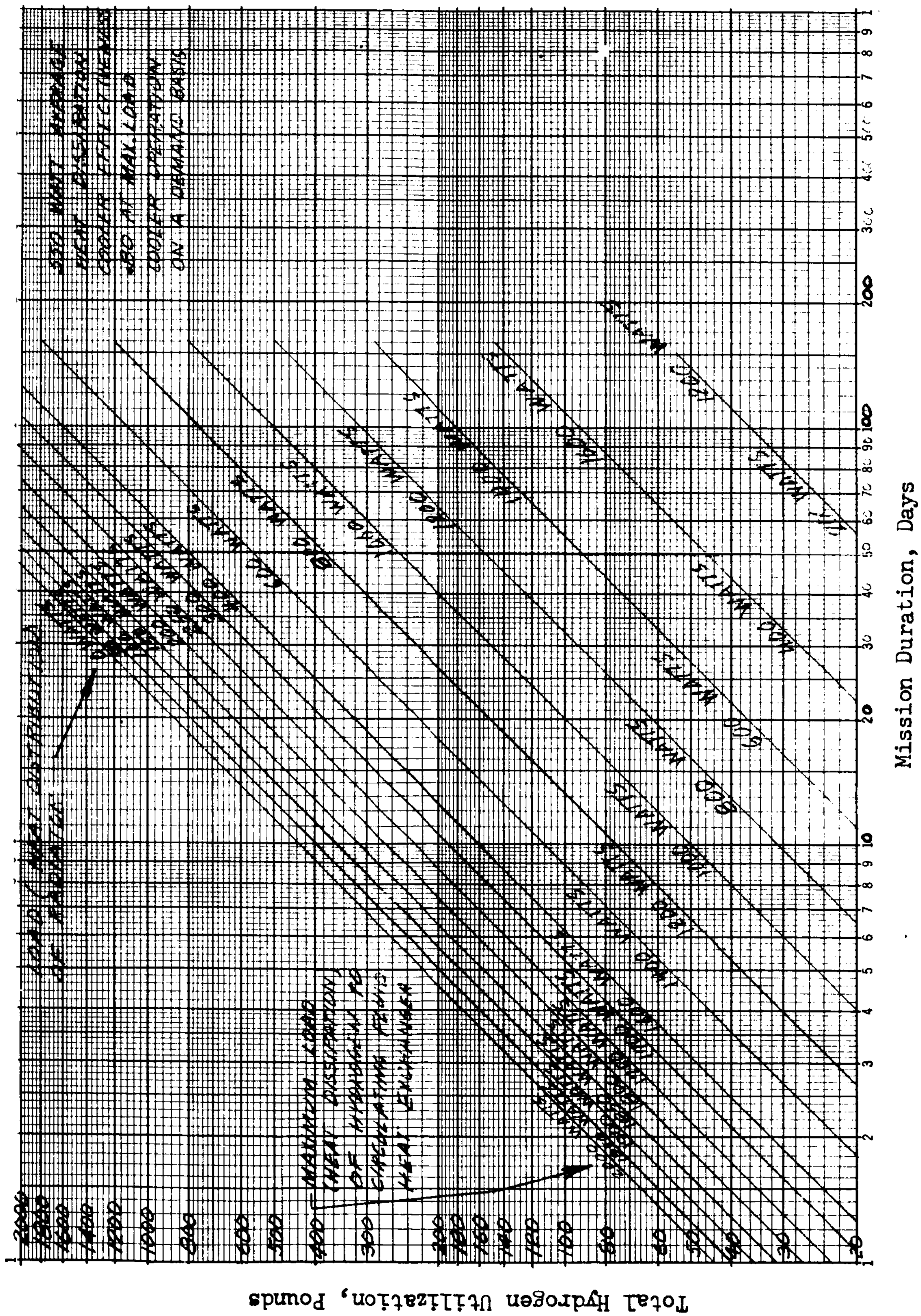


Figure 44. Radiator and Cooler Size as a Function of Hydrogen Utilization, 550 Watt Load

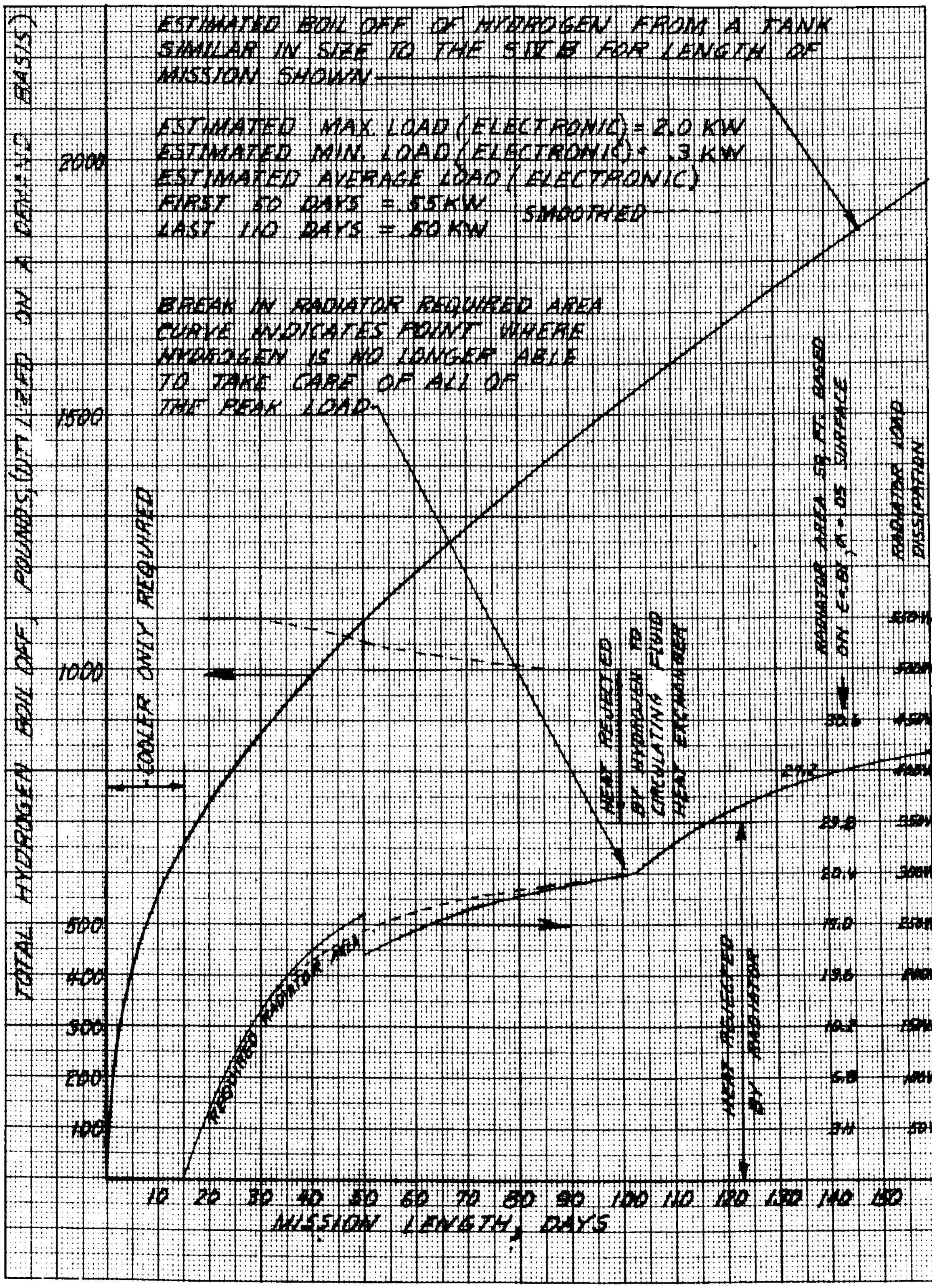


Figure 45. Radiator Area Requirement Versus Mission Duration for Specified Hydrogen Boil-off

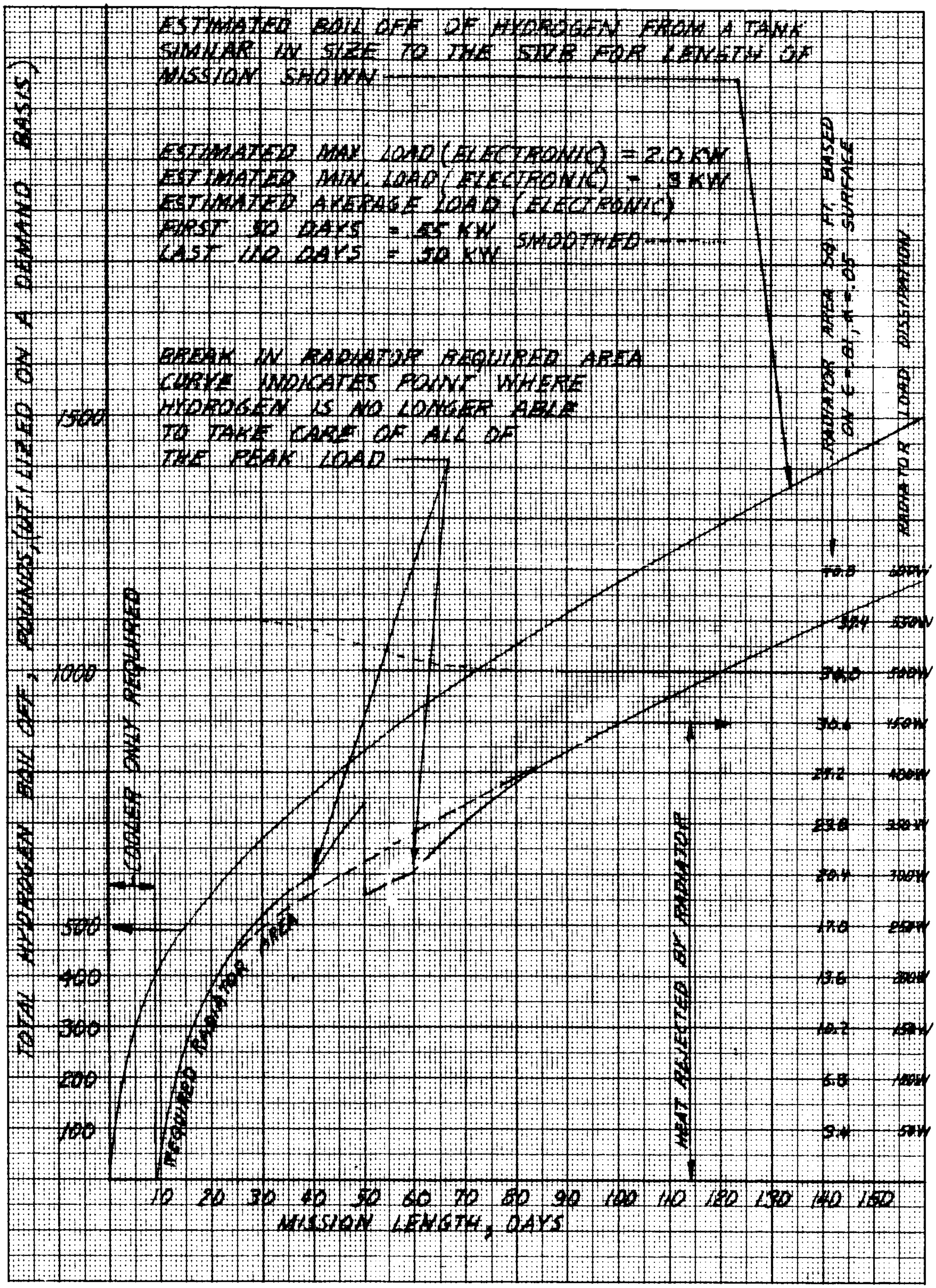


Figure 46. Radiator Area Requirement Versus Mission Duration for Specified Hydrogen Boil-off

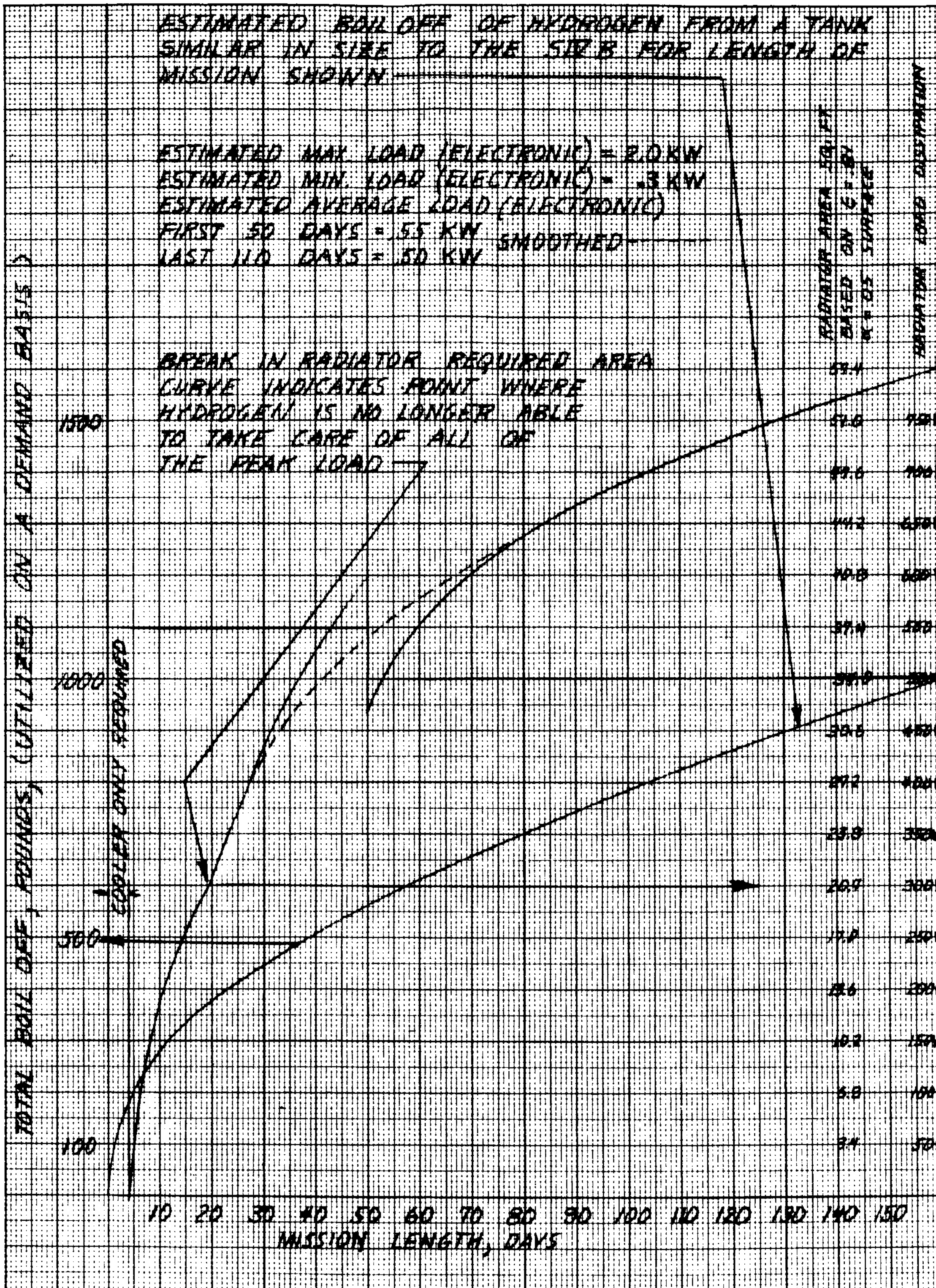


Figure 47. Radiator Area Requirement Versus Mission Duration for Specified Hydrogen Boil-off

1. The effectiveness should be approximately 0.8.

2. The glycol pressure drop should be approximately 0.5 psi.

3. Able to be used with hydrogen subcooled to 30°R.

4. For use with an Ethylene Glycol - 60% water solution circulating cooling fluid.

5. Glycol in-temperature, 40°F.

6. Glycol out-temperature, 20°F.

7. Heat transferred, 2 KW (max.).

8. Design hydrogen inlet temperature, 30°R to 80°R, use 42°R.

9. Assumed glycol congealing temperature, -40°F.

10. Wall temperature not to drop below -40°F even when hydrogen coefficient is three times its calculated value.

11. The glycol side of the exchanger must be entirely parent metal.

12. Hydrogen to contain no entrained liquid.

Three possible designs to meet these design criteria are illustrated in Figure 48. A wider range of designs are possible if the design criteria is altered. The weight however should not change significantly due to design variations. An internally finned-single tube heat exchanger to meet the above conditions was calculated (Appendix A) and used as a basis for parameterization.

The weight of a vent hydrogen-to-60% glycol water exchanger can be estimated from the equation:

$$W_W = \frac{5}{3}(KW)\left[\log_{10}(1-\epsilon)\right] \frac{(SFF)^{0.65}}{(\Delta P_L)^{0.2}(P_G)^{0.2}}$$

where:

W_W = wet weight in pounds

(KW) = heat transferred in kilowatts

ϵ = heat exchanger effectiveness

SSF = system safety factor for freezing of liquid (-40°F) safety factor applied to fluid coefficients

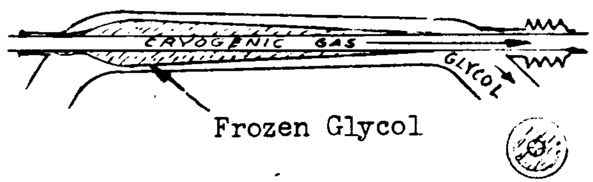
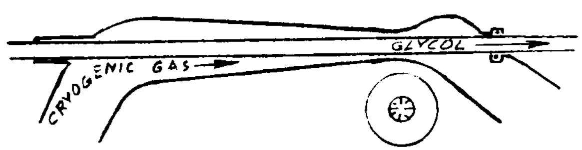
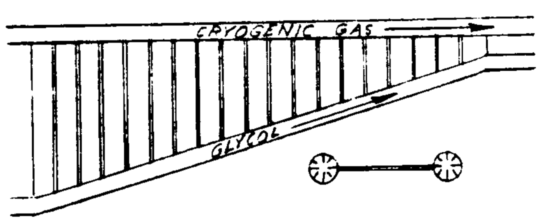
Figures	Description	Region of Usefulness
 Single tube, circulating fluid heat exchanger with controlled freeze up	Freeze up of glycol reduces flow area and increases glycol heat transfer coefficient and area until a balance is achieved. It has a variable pressure drop.	Peak loads of short duration Insensitive to gas temperature
 Single tube, circulating fluid heat exchanger with gas heat transfer coefficient to prevent freezing	Heat transfer coefficients fixed to keep wall temperature above freezing under all possible conditions. Any freeze up is self-controlling in the same way as previous heat exchanger.	Peak loads of short duration Minimum weight (A typical design analysis for H_2-to-glycol heat exchanger is given in the appendix.)
 Single tube, connecting fin, gas to circulating fluid heat exchanger	Fin efficiency controlled heat transfer keeps wall temperature above freezing under all possible conditions. Any freeze up is self-controlling.	Peak loads of short duration Minimum cost Insensitive to gas temperature

Figure 48. Glycol Cooler Concepts

ΔP_L = pressure drop liquid side, psi

P_G = pressure of gaseous hydrogen at exchanger entrance, psia

The above equation is applicable for conditions for which the terms do not differ too greatly from the following:

W_w = 4.0

KW = 2.0

ϵ = 0.78

SSF = 3.0

ΔP_L = 0.50

P_G = 3.5

T_G = 40°F -20°F

A graphical presentation of the above equation is given in Figure 49.

Since two exchangers could handle twice the load (KW), the weight is roughly proportional to the cooling load (i.e., KW appears in the numerator to the 1st power). If an assumption for heat transfer coefficients, independent of area, is made, then the weight is proportional to area which in turn is proportional to NTU's which (for C_{MIN}/C_{MAX} = .2 and .7) is proportional to $-LOG_{10}$ $(1-\epsilon)$. With a lower safety factor, there can be higher hydrogen coefficients in the colder 3/4ths of the exchanger. Though difficult to parameterize, this appears to be reasonably correlated by $(SSF)^{0.65}$. Higher pressure drops can be utilized for greater flow and heat transfer coefficients. Increased coefficients and resultant lower areas appear to be reasonably well correlated by the inverse 0.2 power of the hydrogen and glycol pressure drops.

2.3.4 Space Radiator Design

For analyzing a space radiator, the mathematical equations and parametric data presented in Reference 13, pages 92 through 95, are adequate. These equations and parametric data are useful for sizing a radiator on a preliminary basis which can then be analyzed in detail for the steady-state and transient cases by the use of a computer program.

For a rectangular radiator configuration with multi-tubes or ducts and radiating fins of rectangular cross-section, the following

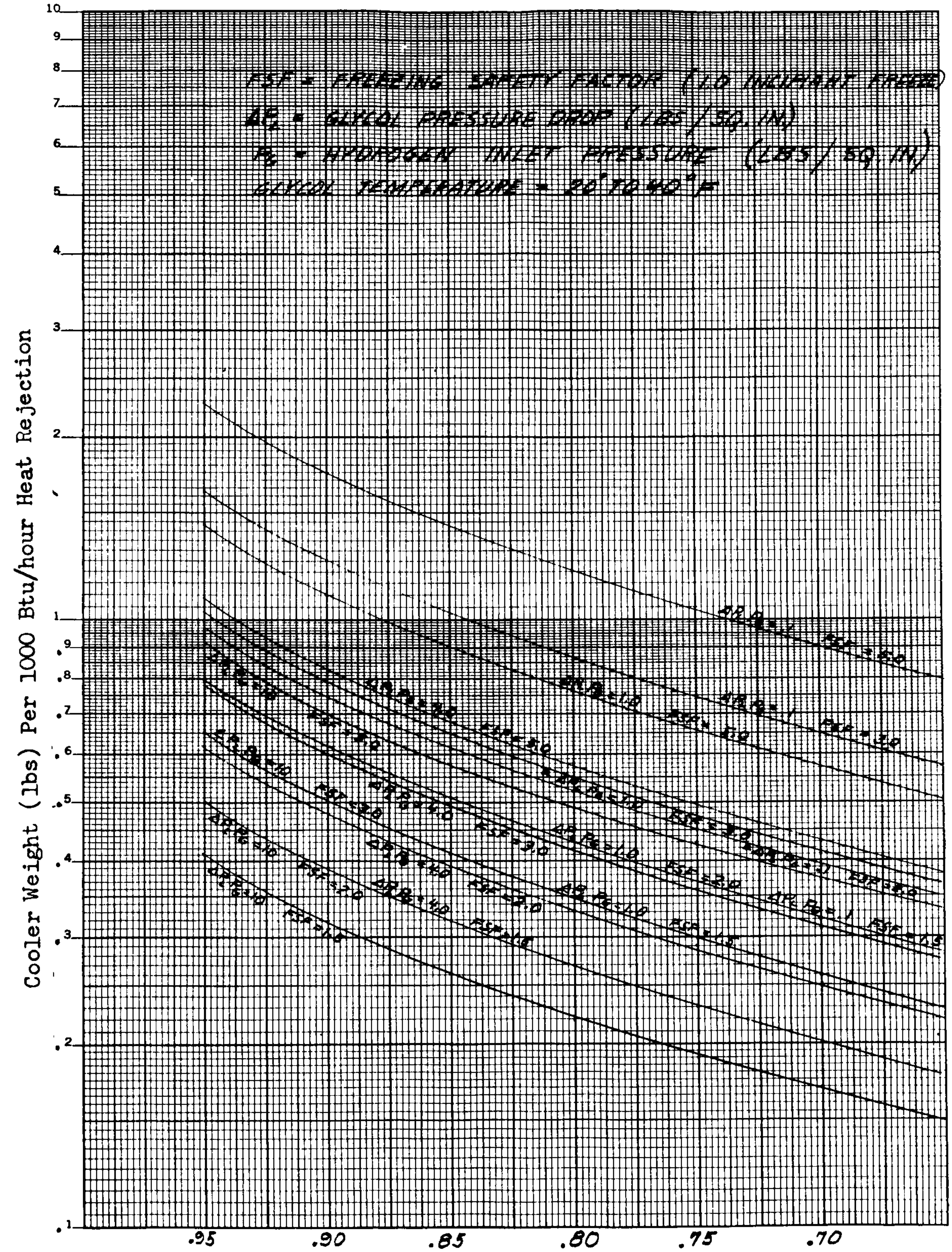


Figure 49. Unit Weight of Hydrogen-to-Glycol Cooler

equations and parametric plots from Reference 13 may be used for either analyzing or sizing a radiator. The length of the radiator as a function of the coolant flow, inlet and outlet temperatures, and other radiator dimensions can be determined from Equation 149 of Reference 13.

$$L_W = \frac{w c_p}{p h} \psi_f + \frac{w c_p}{C_1 T_{W_1}^3 L_{e_{ave}}} \psi_r \qquad (1)$$

where:

$$L_e = \left[L_d + \frac{2 \Omega L_h}{1 - \frac{C_2}{C_1 T_W^2}} \right] F_r \qquad (2)$$

$$\psi_f = - \int_{T_{W_1}}^{T_{W_2}} \frac{4 C_1 T_W^3 \, dT_W}{C_1 T_W^4 - C_2} \qquad (3)$$

$$\psi_r = - C_1 T_{W_1}^3 \int_{T_{W_1}}^{T_{W_2}} \frac{dT_W}{C_1 T_W^4 - C_2} \qquad (4)$$

$$T_{W_2} > \sqrt[4]{\frac{C_2}{C_1}} \qquad (5)$$

The terms ψ_f and ψ_r, can be readily determined from Figure 50 and 51, which are reproduced from Reference 13. The fin effectiveness (Ω) in the equation for the effective width L_e, can be readily determined from Figure 52, which is also reproduced from Reference 13.

The optimum configuration can be determined by an inerative procedure in which the above equations are used to solve for the radiator length L_W for various values for the fin thickness and fin length. For each set of radiator dimensions, the weight-to-heat transfer ratio is determined and this data can be plotted as illustrated in Figure 53.

(See Reference 13 for the description of the nomenclature in the above equations).

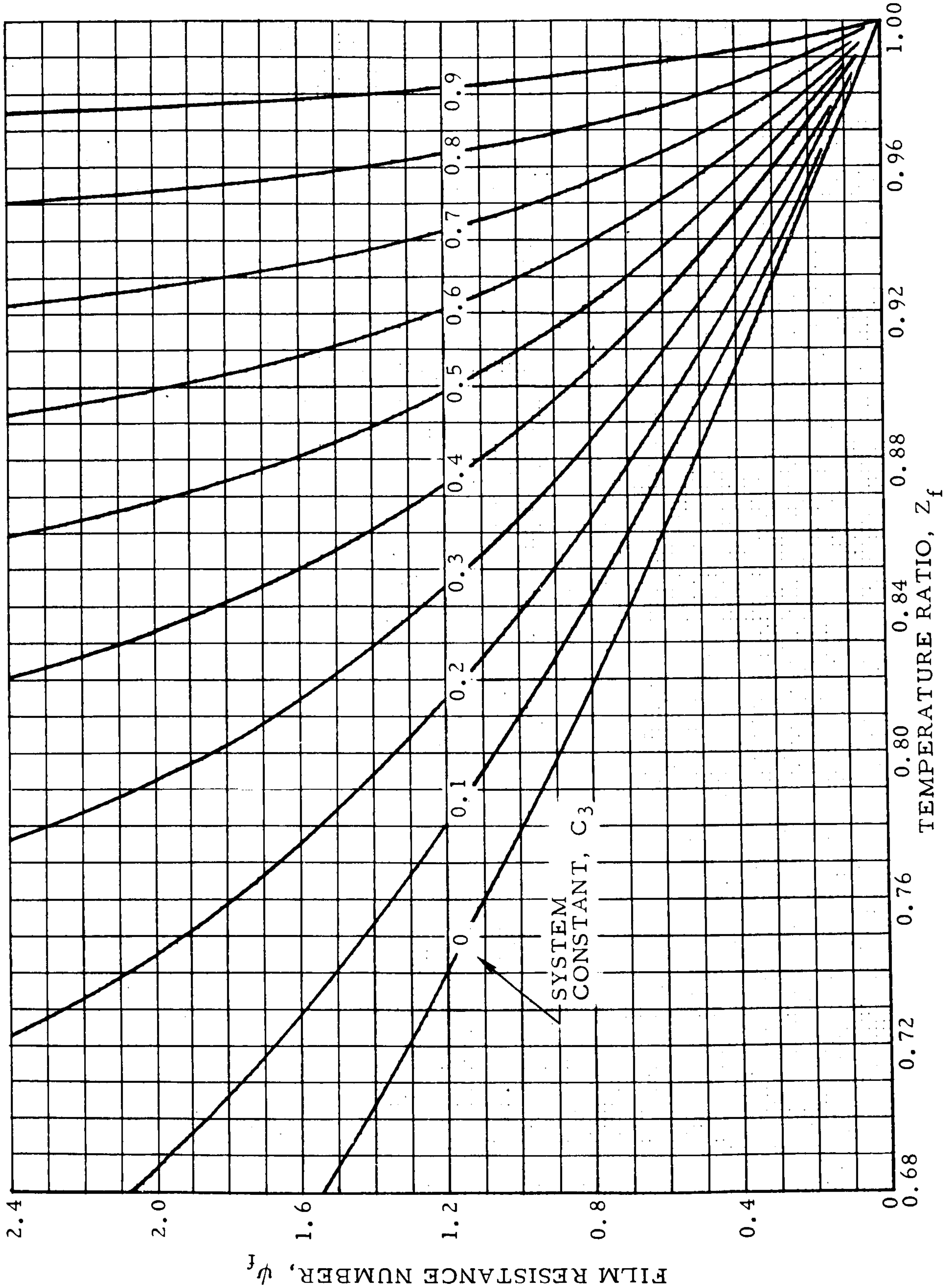


Figure 50. Film Resistance Number for Cooling Processes

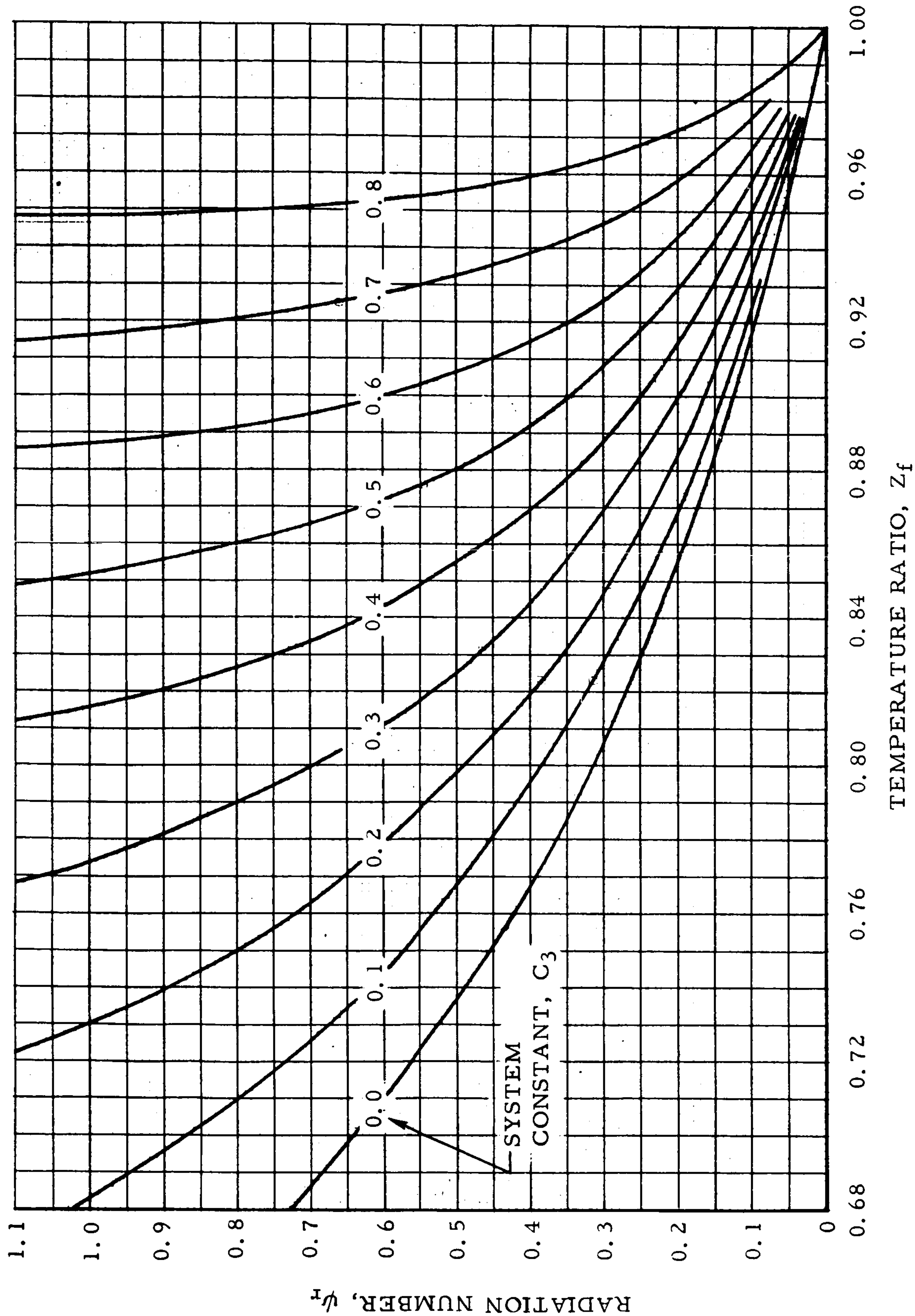


Figure 51. Radiation Number for Cooling Processes

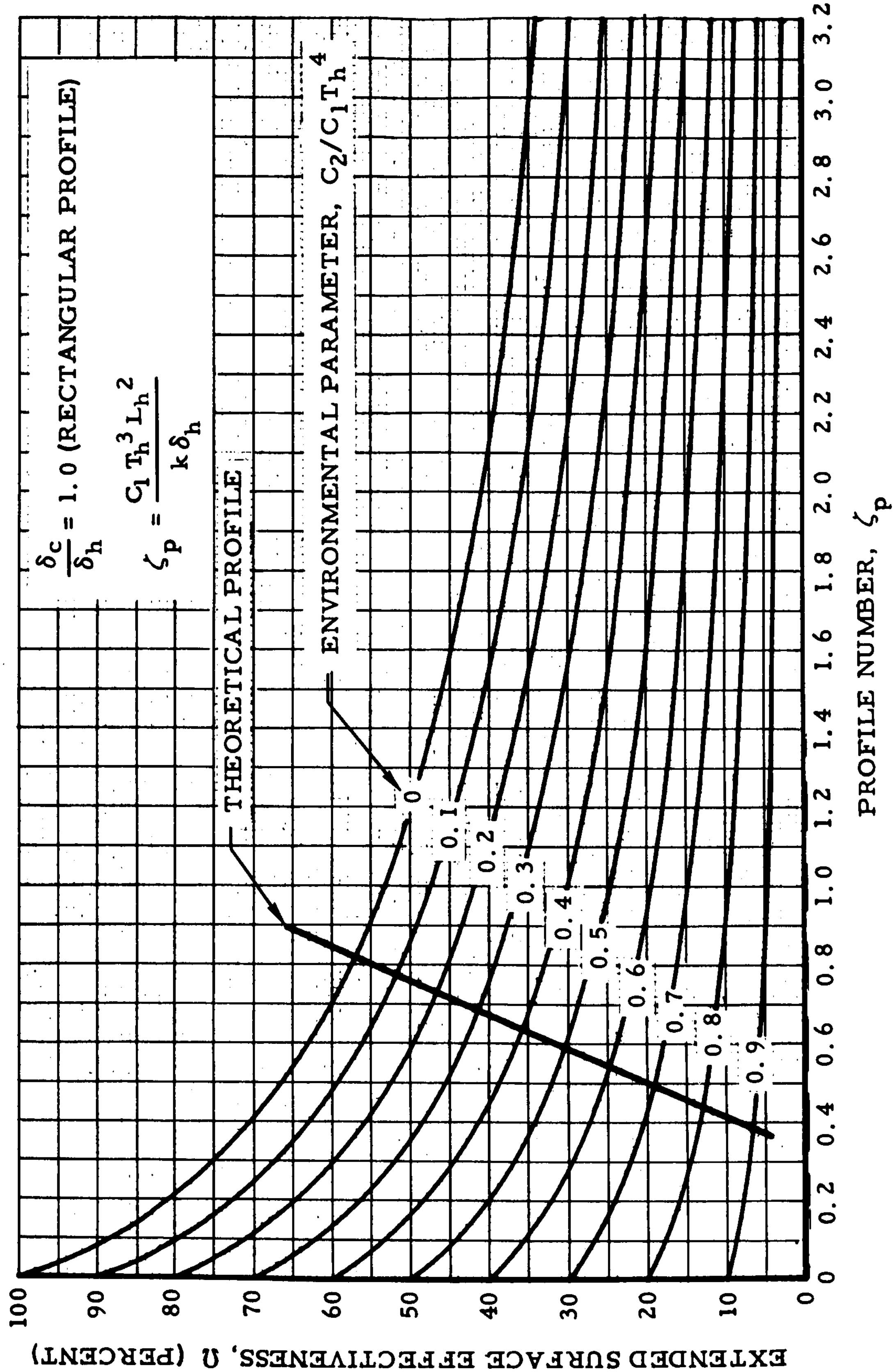


Figure 52. Plate Effectiveness Versus Profile Number for Rectangular Fin

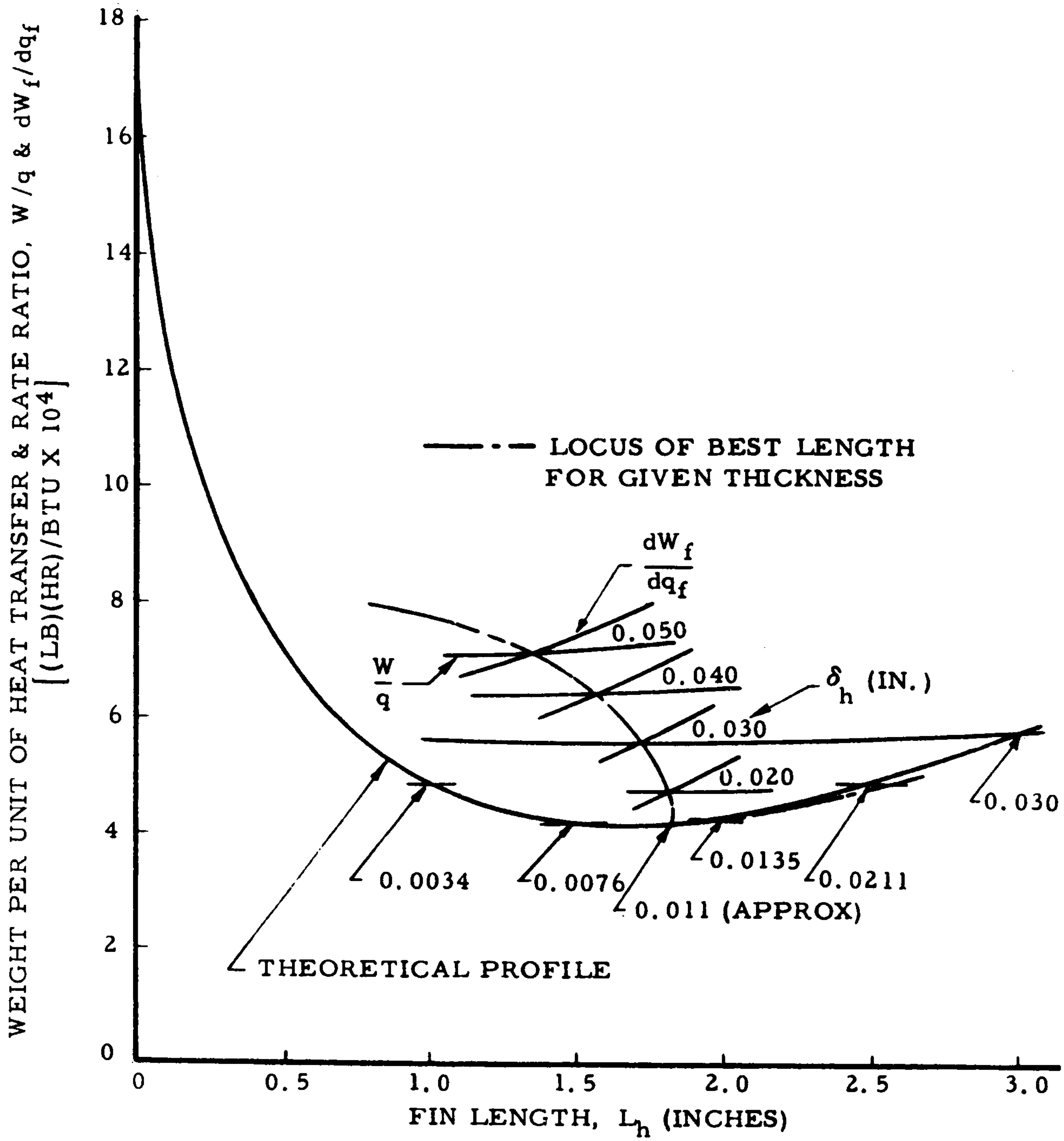


Figure 53. Relationship of Weight Per Unit of Heat Transfer and Rate of Change of Ratio to Fin Length

As shown in Figure 53, the optimum fin length for various fin thickness can be readily established. However, the optimum-weight configuration giving the best ratio of weight-to-heat transfer may not be the most practicable configuration. To establish a practicable configuration, the fin thickness, fin length and radiator length must be taken into consideration.

For example, in Figure 53, the weight-to-heat transfer curve for a fin thickness of 0.030 inches is relatively flat for a range of values of fin length. This means that for a given fin thickness, there are possibly several combinations of fin length and radiator length that give a near optimum configuration.

Because the method for designing or sizing the space radiator is complex, a shorter method for making rough approximations is suggested in Reference 13. This involves neglecting the term ψ_f which is generally smaller than the term ψ_r in Equation 1. This approximation is equivalent to neglecting the temperature differential between the coolant fluid and the tube wall. In most cases, the temperature differential is small and this is illustrated in Figure 54. Figure 54 may also be used in conjunction with Equation 1 in which the tube wall temperature may be established from the coolant temperatures, since in most cases the coolant temperatures are given or specified.

As an aid in the designing or sizing of a space radiator, Table 10 summarizes the flow equation which may be used to establish the radiator duct or tube size for either laminar or turbulent flow conditions. Although the equations are developed for a circular duct, they can be readily modified for semi-circular or similar cross-sectional shape. These equations may also be used to aid in the selection of the coolant fluid. It is to be noted that these equations were developed for both the single tube and multi-tube configurations. The number of tubes or ducts is one of the initial values that must be assumed or defined in order to proceed with the design analysis.

In Table 10, the equations for the heat transfer coefficient for both the laminar and turbulent flow have been arbitrarily selected. Other possible equations which may be used are summarized in Table 11.

Since the radiator weight depends greatly on the radiator material, it is important that the material used for the radiator be carefully selected. Perhaps the most significant criteria for selection are the heat transfer-to-weight ratio, probability of no penetration-to-weight ratio and buckling strength-to-weight ratio. These are summarized in Table 12. On the basis of data currently available, perhaps aluminum is the best choice among the readily available structural materials.

The influence of the structural design and fabrication detail are considered beyond the scope of this study.

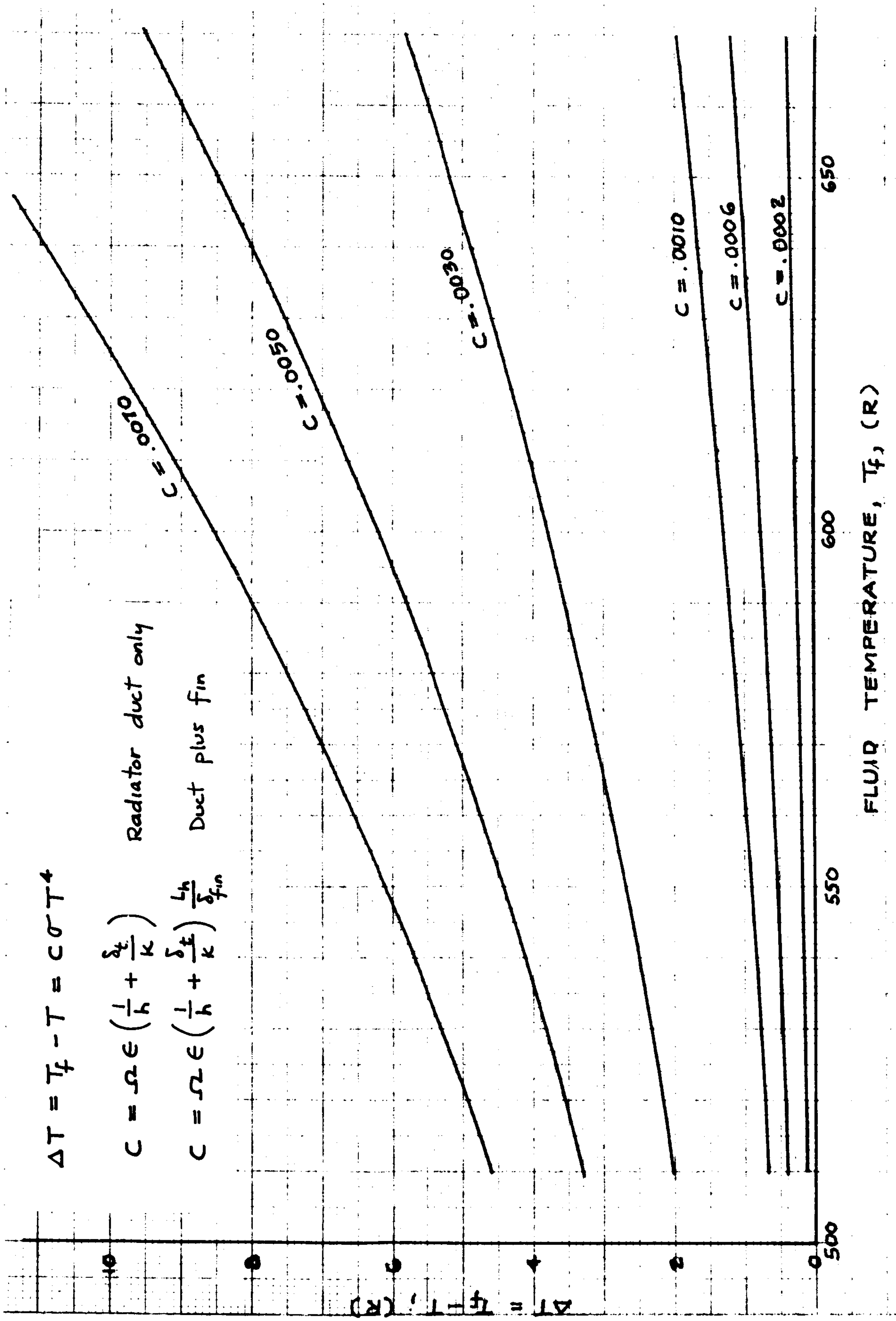


Figure 54. Temperature Differential Versus Fluid Temperature

Table 10. Summary of Heat Transfer and Flow Equations (Round Tube)

	LAMINAR FLOW ($N_R < 2100$)	TURBULENT FLOW ($N_R > 4000$)
1. Pressure drop	$\Delta p = f\rho \frac{L}{D}\frac{V^2}{2g} = \frac{fLW^2}{2g\rho DA^2}$	$\Delta p = f\rho \frac{L}{D}\frac{V^2}{2g} = \frac{fLW^2}{2g\rho DA^2}$
2. Friction factor (Darcy-Weisbach)	$f = \frac{64}{N_R} = \frac{64\mu A}{DW}$	$f = \frac{0.3164}{(N_R)^{0.25}} = 0.3164\left(\frac{\mu A}{DW}\right)^{0.25}$
3. Pressure drop per tube	$\Delta p_1 = \frac{32\mu}{g\rho}\frac{W_o}{D_o^2 A_o} L_1 n = \left(\frac{128}{\pi g}\right)\left(\frac{\mu}{\rho}\right)\left(\frac{W_o}{D_o^4}\right)(L_1 n)$	$\Delta p_1 = \frac{0.1582\, W_o^{1.75}\mu^{0.25}}{g\rho\left(\frac{\pi}{4}\right)^{1.75} D_o^{4.75}} L_1 n^{0.625}$ $= \left(\frac{0.1582}{g}\right)\left(\frac{4}{\pi}\right)^{1.75}\left(\frac{\mu^{0.25}}{\rho}\right)\left(\frac{W_o^{1.75}}{D_o^{4.75}}\right)(L_1 n^{0.625})$
4. Pump power	$\frac{W_1 \Delta p_1}{\rho} = \left(\frac{32}{\pi g}\right)\left(\frac{\mu}{\rho^2 C_p^2}\right)\left(\frac{q}{\Delta T D_o^2}\right)^2 L_1$	$\frac{W_1 \Delta p_1}{\rho} = \left(\frac{0.1582}{g}\right)\left(\frac{4}{\pi}\right)^{1.75}\left(\frac{q}{\Delta T}\right)^{2.75}\left(\frac{1}{D_o^{4.75}}\right)\left(\frac{\mu^{0.25}}{\rho^2 C_p^{2.75}}\right)\left(\frac{L_1}{n^{0.875}}\right)$
5. Heat transfer coefficient	$h = 1.86\frac{K}{D}\left[N_R \frac{C_p\mu}{K}\frac{L}{D}\right]^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0.14}$	$h = 0.023\frac{K}{D}(N_R)^{0.8}\left(\frac{C_p\mu}{K}\right)^{0.4}$
6. Heat transfer coefficient for individual tube	$h_1 = 1.86\frac{k^{2/3}}{D_o}\left[\left(\frac{q}{\Delta T}\right)\left(\frac{4}{\pi}\right)\left(\frac{L_1}{D_o^2}\right)\right]^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0.14}\sqrt{n}$ $= 1.86\left[\left(\frac{q}{\Delta T}\right)\left(\frac{4}{\pi}\right)\right]^{1/3}\left(\frac{1}{D_o^{1.67}}\right)(K^{2/3})\left(\frac{\mu_b}{\mu_w}\right)^{0.14}(L_1^{1/3})\sqrt{n}$	$h_1 = 0.023\frac{k}{D_o}\left(\frac{D_o W_o}{A_o \mu}\right)^{0.80}\left(\frac{C_p\mu}{k}\right)^{0.40}(n)^{0.10}$ $= 0.023\left[\left(\frac{q}{\Delta T}\right)\left(\frac{4}{\pi}\right)\right]^{0.80}\left(\frac{1}{D_o^{1.80}}\right)\left(\frac{k^{0.60}}{C_p^{0.40}\mu^{0.40}}\right)(n)^{0.10}$
7. $\frac{\text{Radiator pump power}}{\text{Radiator heat rejection}}$	$\frac{W_1 \Delta p_1}{n\rho h_1 L_1 D_1 \Delta T} = \left(\frac{32}{1.86 g\pi}\right)\left(\frac{\pi}{4}\right)^{1/3}\left(\frac{1}{\Delta T}\right)\left(\frac{q}{\Delta T}\right)^{5/3}\left(\frac{\mu}{K^{5/3}\rho^2 C_p^2}\right)\left(\frac{\mu_b}{\mu_w}\right)^{0.14}\left(\frac{1}{D_o^{11/3}}\right)\left(\frac{1}{nL_1^{1/3}}\right)$	$\frac{W_1 \Delta p_1}{n\rho h_1 L_1 D_1 \Delta T} = \frac{0.1582}{0.023\, g\left(\frac{\pi}{4}\right)^{0.95}}\left(\frac{q}{\Delta T}\right)^{1.95}\left(\frac{1}{\Delta T}\right)\left(\frac{\mu^{0.65}}{\rho^2 k^{0.60} C_p^{2.35}}\right)\left(\frac{1}{D_o^{3.95}}\right)\left(\frac{1}{n^{0.975}}\right)$

A - cross-section area, sq ft
C_p - fluid heat capacity, Btu/(lb)(F)
D - tube diameter, ft.
f - friction faction
g - gravitational constant, 32.2 ft/sec²
h - heat transfer coefficient, Btu/(hr)(sq ft)(F)
K - fluid thermal conductivity, Btu/(hr) sq ft)(F)
L - tube length, ft
N_R - Reynolds number
n - number of radiator tubes
Δp - pressure drop, psf
q - heat flow, Btu/sec

ΔT - temperature differential, F
v - flow velocity, ft/sec
W - flow rate, lb/sec

ρ - fluid density, lb/cu ft
μ - fluid viscosity, lb/(ft)(sec)

Subscript
o - single tube basis
1 - multi-tube basis
b - bulk temperature condition
w - wall temperature condition

Table 11. Equations for Fluid Heat Transfer Coefficients in Smooth Tubes

Flow	Equation*	Reference
Laminar Re<2100	(1) $Nu_b = 1.86 (\phi_b)^{1/3} (\mu_b/\mu_w)^{0.14}$ where $\phi_b = (RePr)\ D/L$ $\phi > 20$	Sieder-Tate (Jacob, Vol. I) Reference 14
	(2) $Nu_m = 3.65 + \frac{0.0668\phi}{1+0.04(\phi)^{2/3}}$ $\phi \leq 20$	Nusselt-Graetz (Eckert and Drake) Reference 15
Turbulent Re>2100	(3) $Nu_m = 0.023 (Re)^{0.8} (Pr)^{1/3}$ $400 \leq L/D$	Colburn (Giedt) Reference 16
	(4) $Nu_b = 0.023 (Re)^{0.8} (Pr)^{1/3} (\mu_b/\mu_w)^{0.14}$	Sieder-Tate (McAdams) Reference 17
	(5) $Nu_b = 0.036 (Re)^{0.8} (Pr)^{1/3} (\mu_b/\mu_w)^{0.14} (D/L)^{0.1}$ $10 < L/D < 400$	Cholette (Jacob, Vol. I) Reference 14

*Subscripts:
b — Properties evaluated at average fluid bulk temperature $\left(t_b = \frac{t_{IN} + t_{OUT}}{2}\right)$
m — Properties evaluated at mean film temperature $\left(t_m = \frac{t_{FLUID\ AVG} + t_{WALL\ AVG}}{2}\right)$

Table 12. Summary of Equations for Material Selection

Heat Transfer (Conduction)	$q = K A \Delta T/\delta$ (Fourier Eq.)	Heat transfer-to-weight ratio $\propto k/\rho$
Probability of no meteoroid penetration	$P_{(0)} = e^{-\frac{K^{10/3} v^{10/9} A\theta}{10^{12} \delta^{10/3}}}$ (Bjork Eq.)	Probability of no penetration-to-weight ratio $\propto 1/\rho K$
Buckling Strength	$F_{cr_c} = \eta C E_c \delta/r$	Buckling strength-to-weight ratio $\propto E_c/\rho$

Weight Analysis

Since the radiator may be a major contributor to the total weight of a recycle cooling system such as a closed loop liquid system, a weight analysis was initiated to determine the percentage contribution of the radiator elements, to establish a probable range of radiator weight per unit area, and to determine the influence of material densities. This information should be helpful in making preliminary design estimates and to know the sensitivity of the radiator weight to various factors.

One radiator configuration has been investigated which is illustrated in Figure 55. It is considered to be a practical configuration which can be either an integral part of a vehicle structure or as a separate, mountable panel. Also, as illustrated in Figure 55, a relatively simple means for providing meteoroid protection can be incorporated into the design. One additional consideration that enters into the design is the ease of fabrication.

For this analysis, the total radiator weight was divided into the fin weight, the tube weight, the fluid holdup weight, and the header weight; the total radiator weight equals the sum of these four components. Table 13 summarizes the equations used in the analysis.

For purposes of the weight analysis, three different cases were analyzed for assumed values for radiator dimensions and material densities and with the additional assumption that all parts of the radiator are of the same material. Table 14 gives the dimensions and densities used in the analysis.

The results of the analysis is given in Table 14. As indicated, the radiator fin is the major contributor to the total weight, nearly two-thirds of the total. Reducing the fin area, fin thickness and material density are the obvious ways to reduce the overall radiator weight. The fluid holdup is the next major contributor. A fifty percent reduction in fluid density will reduce the overall weight by about ten percent. Thus, some weight reduction may be possible by a combination of a reduction in fluid density and the amount of fluid holdup. The radiator tubes and the headers make a small contribution to the total weight and thus the pressure drop and flow conditions should dictate the size of the tubes or flow passages.

For rapid approximation of a radiator weight, a specific weight of about one pound per square foot (1 lb/sq ft) appears to be reasonable based on the assumed material densities.

From the foregoing, it is obvious that significant weight savings can be achieved by considering an integral radiator and vehicle structure, even at the expense of requiring a larger radiating surface to accommodate possible unfavorable orientation of the orientation of the radiating surface with respect to the incident heat loads.

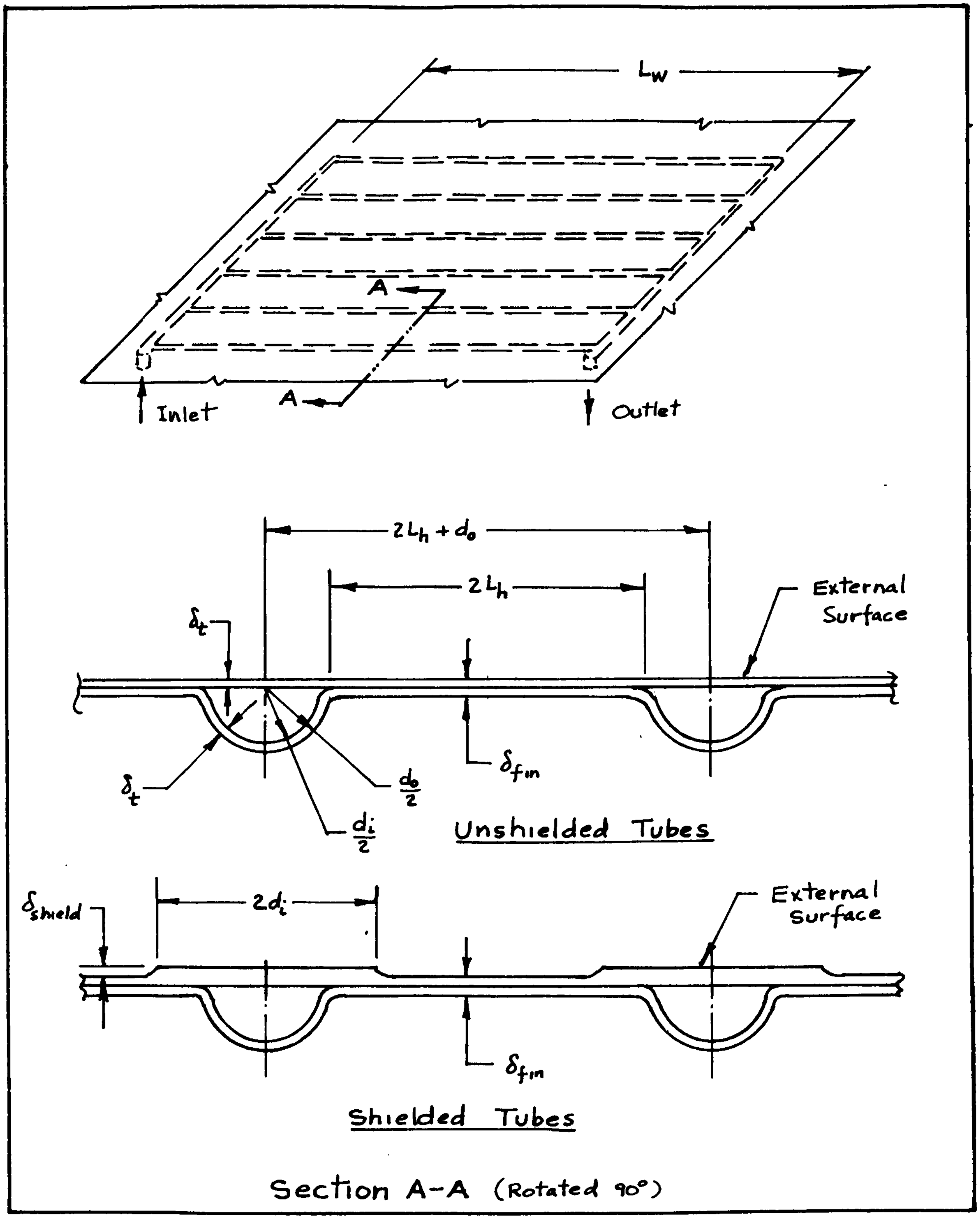


Figure 55. Rectangular Radiator Configuration

Table 13. Summary of Weight Equations

$$W_{Total} = W_{Fins} + W_{Tubes} + W_{Fluid} + W_{Headers}$$

$$W_{Fins} = n L_W \rho_{fin} \left(2 L_h \delta_{fin} \right)$$

$$W_{Tubes} = n L_W \rho_{fin} \left\{ \frac{\pi}{8} d_i^2 \left[\left(1 + \frac{2\delta_t}{d_i}\right)^2 - 1 \right] + d_i \delta_t \left(1 + \frac{2\delta_t}{d_i}\right) \right\}$$

$$W_{Fluid} = n L_W \rho_{fluid} \frac{\pi}{4} d_i^2 \left\{ \frac{1}{2} + \frac{n}{L_W} \left[d_i \left(1 + \frac{2\delta_t}{d_i}\right) + 2 L_h \frac{n-1}{n} \right] \right\}$$

$$W_{Header} = n L_W \rho_{fin} \frac{\pi}{4} d_i^2 \left\{ \frac{n}{L_W} \left[d_i \left(1 + \frac{2\delta_t}{d_i}\right) + 2 L_h \frac{n-1}{n} \right] \left[\left(1 + \frac{2\delta_t}{d\sqrt{n}}\right)^2 - 1 + \frac{8\delta_t}{\pi d_i \sqrt{n}} \left(1 + \frac{2\delta_t}{d_i \sqrt{n}}\right) \right] \right\}$$

$$\frac{W_{Total}}{n L_W} \frac{1}{\rho_{fin}} = 2 L_h \delta_{fin} + d_i \delta_t \left(1 + \frac{2\delta_t}{d_i}\right) + \frac{\pi}{4} d_i^2 \left\{ \frac{1}{2} \left[\frac{\rho_{fluid}}{\rho_{fin}} + \left(1 + \frac{2\delta_t}{d_i}\right)^2 - 1 \right] \right.$$

$$\left. + \frac{n}{L_W} \left[d_i \left(1 + \frac{2\delta_t}{d_i}\right) + 2 L_h \frac{n-1}{n} \right] \left[\frac{\rho_{fluid}}{\rho_{fin}} + \left(1 + \frac{2\delta_t}{d_i \sqrt{n}}\right)^2 - 1 + \frac{8}{\pi} \left(\frac{\delta_t}{d_i \sqrt{n}}\right) \left(1 + \frac{2\delta_t}{d_i \sqrt{n}}\right) \right] \right\}$$

Table 14. Space Radiator Weight Summary

	A	B	C
n	5.0	10	10
δ_{fin}, ft	0.0040	0.0040	0.005
L_h, ft	0.0833	0.2500	0.250
d_i, ft	0.0250	0.0500	0.050
δ_t, ft	0.0020	0.0020	0.0025
L_w, ft	5.000	10.000	10.000
ρ_{fluid}, lb/cu ft	60.000	60.000	60.000
ρ_{fin}, lb/cu ft	150.000	150.000	150.000
Fin Weight, W_{fin}, lb	2.50	30.00	37.50
Tube Weight, W_{tube}, lb	.54	4.10	5.16
Fluid Weight, W_{fluid}, lb	.49	11.88	11.88
Header Weight, W_{header}, lb	.07	1.25	1.60
Total Weight, lb	3.595	47.223	56.14
W_{fin}/W_{total}, %	69.50	63.50	66.70
W_{tube}/W_{total}, %	14.88	8.68	9.20
W_{fluid}/W_{total}, %	13.52	25.11	21.12
W_{header}/W_{total}, %	2.04	2.64	2.85
Radiator Area, sq ft	4.16	50.00	50.00
Radiator Weight/Area, lb/sq ft	0.865	0.945	1.123

The above analysis did not include the influence of a meteoroid armour such as a relatively simple one indicated in Figure 55. A weight analysis of the simple armour, which consisted of doubling the thickness of the external, flat surface of the tubes, indicates a possible five to ten percent increase in the total radiator weight. This suggests that the addition of a meteoroid armour would be highly desirable if a significant increase in probability-of-no puncture is possible. This will be investigated further as part of the system reliability-weight trade-off.

α/ϵ Ratio Selection

As a continuation of the work described in Section 2.3.4 of the first quarterly report, Reference 1, the effect of surface coating (α/ϵ) on equilibrium temperature of a flat radiator located on a vehicle surface which is perpendicular to the earth surface and in a circular earth orbit at a height of 400 nautical miles was analyzed. For the analysis, an equilibrium heat balance equation was used in which the terms were rearranged to solve for the α/ϵ ratio:

$$\alpha/\epsilon = \frac{1}{Q_S + Q_{SR}} \left[\sigma T^4 - Q_{EE} - Q_{int}/\epsilon A\right]$$

$$\alpha/\epsilon = \frac{1}{C_1} \left[\sigma T^4 - C_2 - Q_{int}/\epsilon A\right]$$

where:

$C_1 = Q_S + Q_{SR}$, Btu/(hr)(sq ft)

$C_2 = Q_{EE}$, Btu/(hr)(sq ft)

Q_{int} = equipment heat load, Btu/hr

T = radiator equilibrium temperature, R

Q_S = direct solar heat, Btu/(hr)(sq ft)

Q_{SR} = reflected solar heat, Btu/(hr)(sq ft)

Q_{ER} = planetary emission, Btu/(hr)(sq ft)

The above equation was used to make the graphic presentation in Figure 56 for the case of maximum environmental heat load for three values of the term $Q_{int}/\epsilon A$. It is to be noted that a family of such

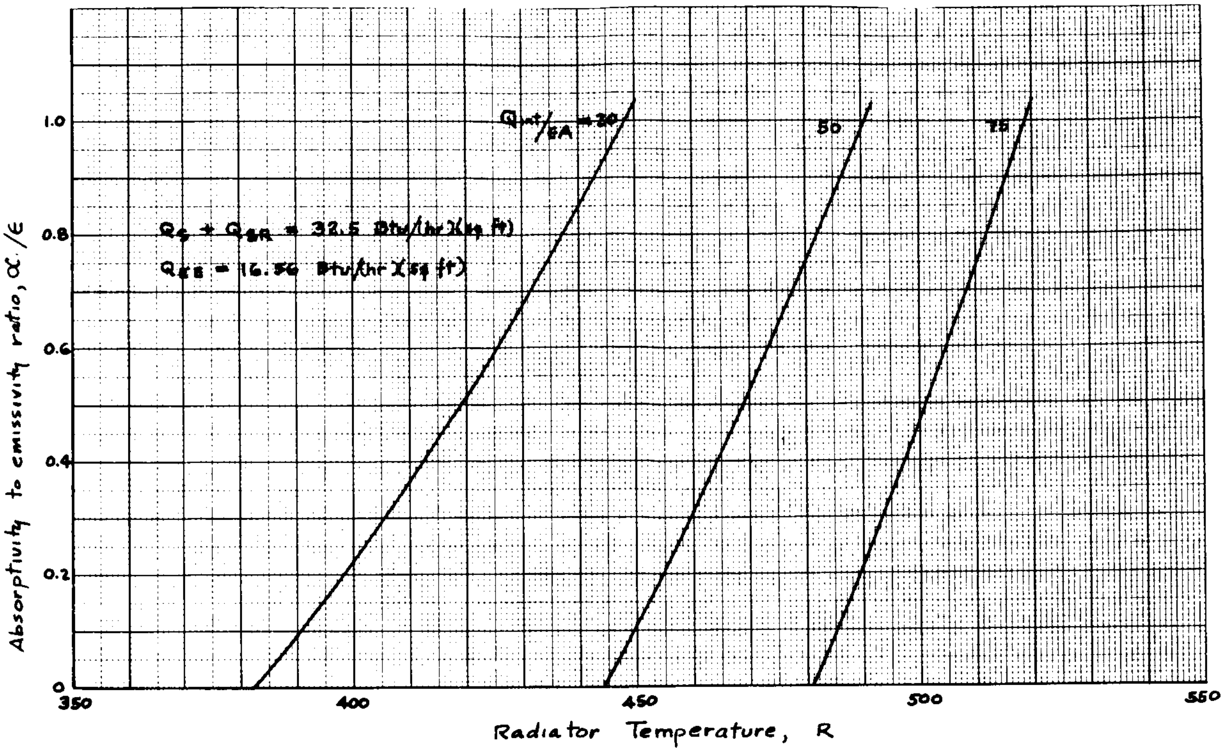


Figure 56. α/ε Ratio Versus Radiator Temperature

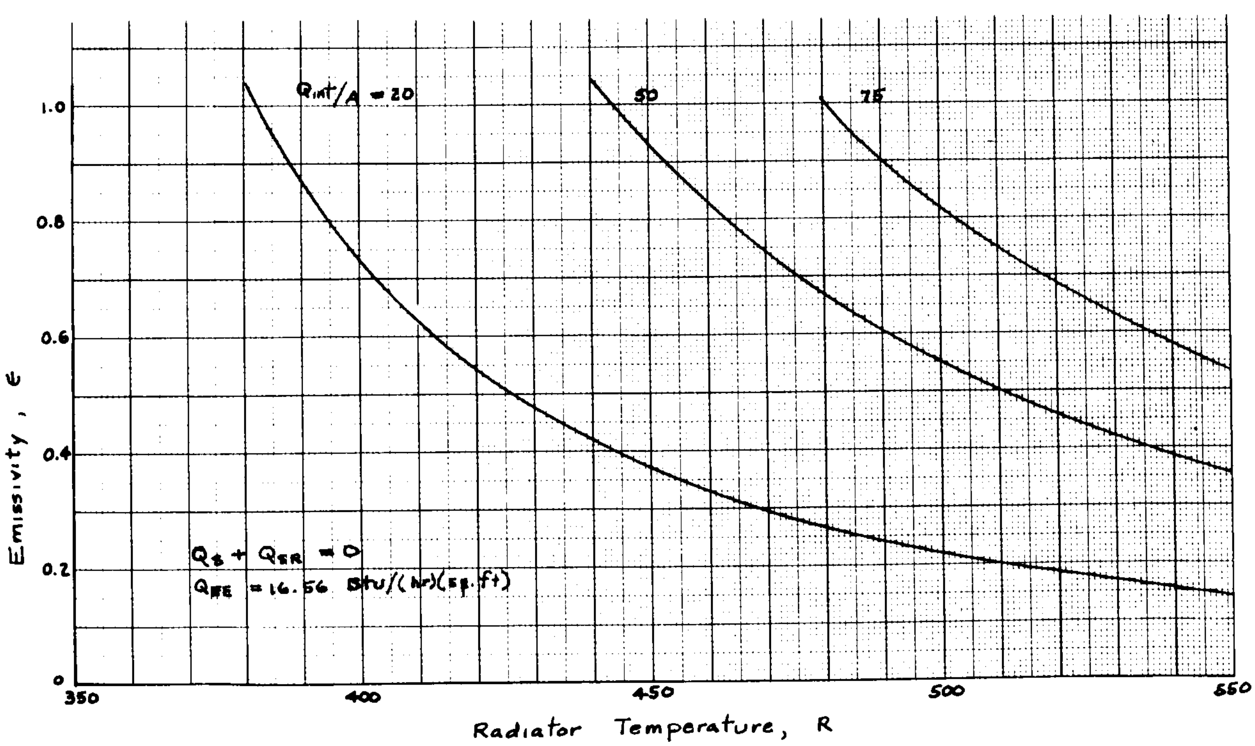


Figure 57. Emissivity Versus Radiator Temperature

curves would be useful for the rapid selection of the ratio α/ϵ, emissivity (ϵ), and the radiator area for a range of equipment heat loads for a particular radiator equilibrium temperature. Another possible use of a plot similar to Figure 56, is in estimating the effect of a change in the ratio α/ϵ for a particular radiator design. The degradation of the surface coating due to space environmental condition could result in the increase in the ratio α/ϵ, and thus as indicated in Figure 56, the radiator temperature would increase for a particular heat load and radiator design. Although a more detailed analysis would be required before a definite conclusion can be made, it does appear that the radiator temperature does not vary appreciably for a relatively large change in the ratio α/ϵ for a given heat load and radiator area.

For the case of minimum environmental heat load, when only the planetary emission is considered, the emissivity and not the ratio is solved from the equilibrium heat balance equation. The resulting equation is:

$$\epsilon = Q_{int} / A \left(\sigma T^4 - C_2 \right)$$

The graphical presentation of this equation is given in Figure 57 for three values of Q_{int}/A. Figures such as Figure 57 together with figures similar to Figure 56 provide a useful means for making rapid, preliminary estimates of radiator performance for varying conditions. For example, the sensitivity of the radiator temperature with changes in the surface coating due to environmental conditions can be readily assessed. The need for compensation for degraded performance can be readily determined.

Equation 4 on Page 79 of the first quarterly report, Reference 1, is in error and should be corrected to read as:

$$\frac{\alpha}{\epsilon} = \frac{\sigma T_s'^4 - \frac{Q_D}{\epsilon A} - \frac{M_R C_{P_R} (T_1 - T_s')}{\epsilon A \Delta t} - (1-\bar{A}_L)\sigma \left(1 - \sqrt{1-\left(\frac{r}{r+h}\right)^2}\right) (T_p^4 - T_s^4) \cos\beta}{\frac{442}{(AU)^2} \left[\cos\theta + \bar{A}_L \left(1 - \sqrt{1-\left(\frac{r}{r+h}\right)^2}\right) \cos\psi \right]}$$

The equation given on Page 80 in the same quarterly report should also be changed to read as:

$$T_s'^4 + \frac{M_R C_{P_R}}{\sigma \epsilon A \Delta t} T_s' = \frac{Q_D}{\sigma \epsilon A} + \frac{M_R C_{P_R} T_1}{\sigma \epsilon A \Delta t}$$

$$- (1 - A_\ell)\left(1 - \sqrt{1 - \left(\frac{r}{r+h}\right)^2}\right)(T_p^4 - T_s^4) \cos\beta$$

2.3.5 Temperature Regulation

There are a number of methods or combination of methods which can be used to achieve the required temperature regulation of a recycle cooling method with a space radiator for heat rejection. Table 15 gives a list of alternate methods for the cases of constant coolant circulation rate and variable coolant circulation rate. Fixed radiator and radiator by-pass methods are briefly discussed in the following paragraphs. These and the others listed in Table 15 will be investigated in more detail during the next quarterly period.

Perhaps the simplest method to achieve temperature regulation is to vary the coolant flow rate to adjust to the variation in the heat rejection rate and/or the external environmental conditions. No radiator by-pass or means for varying the radiator effective area or surface characteristics would be used. The variation in coolant flow rate could be achieved by varying the pump speed or by increasing the line pressure drop or by the combination of the two. The required control elements would be a temperature sensor located at the equipment coolant inlet or outlet or sensors at both the coolant inlet and outlet and a controller to regulate the pump speed or to regulate a restrictor to vary the line pressure drop. The effect of varying the flow rate with variation in heat loads will be to vary the temperature levels from the design points throughout the closed liquid loop and only the temperature differences across the equipment and radiator would remain constant. Since the temperature of the coolant into the equipment is equal to the temperature of the coolant out of the radiator, variations in the heat load will result in variations of coolant temperature into the equipment.

To illustrate the variation in the coolant temperature out of the radiator with variations in the flow rate and the heat loads, the

Table 15. Alternate Methods for Temperature Regulation

$\dot{W}$ (constant)	$\dot{W}$ (variable)
	1. Fixed radiator
1. Radiator by-pass (vary flow to radiator) Diagram (a)	2. Radiator by-pass (vary flow to radiator) Diagram (a)
2. Segmented radiators with by-passes (vary flow and/or radiating area) Diagram (b)	3. Segmented radiators with by-passes (vary flow and/or radiating area) Diagram (b)
3. Vary radiator orientation • movable radiator • vary vehicle attitude	4. Vary radiator orientation • movable radiator • vary vehicle attitude
4. Vary radiator surface characteristics (surface coating)	5. Vary radiator surface characteristics (surface coating)
5. In-line regenerator with by-pass Diagram (c)	6. In-line regenerator with by-pass Diagram (c)
6. In-line heat source • electrical heat • waste heat	7. In-line heat source • electrical heat • waste heat
7. In-line cooler (auxiliary heat sink)	8. In-line cooler (auxiliary heat sink)

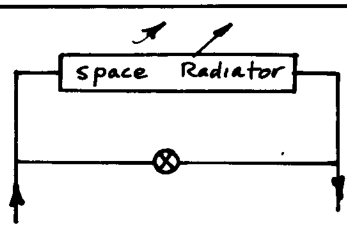


Diagram (a)

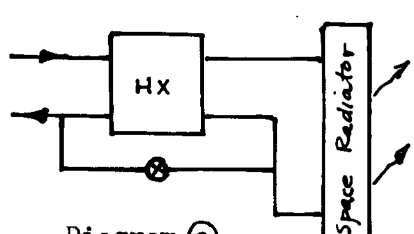
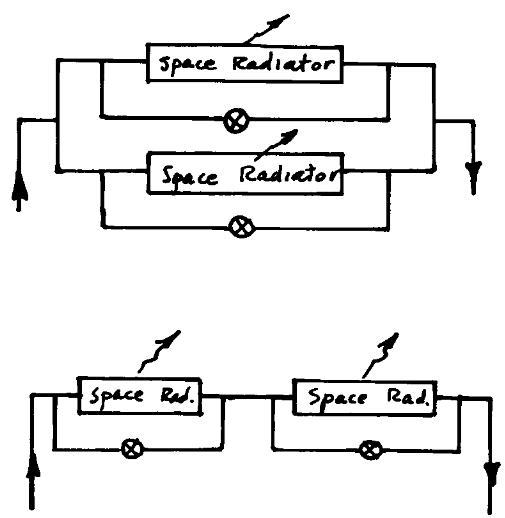


Diagram (c)



Diagrams (b)

following equation was derived for the equilibrium case:

$$t_{r_o} = \sqrt[4]{\frac{q_E}{(\epsilon A)_{des}\,\sigma\,\Omega}} - 460 - \frac{q_E}{W C_p}$$

where:

t_{r_o} = coolant temperature, radiator outlet, F

$$\Omega = \frac{q_{actual}}{(\epsilon A)_{des}\,\sigma\,T_a^4}$$

A = radiator surface area, sq ft

C_p = coolant specific heat, Btu/(lb)(F)

q_E = equipment heat load, Btu/hr

q_{net} = net heat rejected to space, Btu/hr

T_a = coolant temperature, equipment outlet, R

t_a = coolant temperature, equipment outlet, F

W = coolant flow rate, lb/hr

ϵ = radiator surface emissivity

σ = Stefan-Boltzmann constant

By expressing the various terms as a ratio with respect to the design point condition, the above equation becomes:

$$\frac{t_{r_o}}{t_b} = \left(\frac{460 + t_a}{t_b}\right)_{des} \sqrt[4]{\frac{q_E}{q_{E_{des}}}\,\frac{\Omega_{des}}{\Omega}} - \frac{460}{t_b} - \left(\frac{q_E}{q_{E_{des}}}\right)\left(\frac{W_{des}}{W}\right)\left(\frac{t_a}{t_b} - 1\right)_{des}$$

where:

t_b = design point coolant temperature, radiator outlet, F

subscript des = design point

Figure 58 is a graphic presentation of the above equation in which the following values were assumed:

$$\left(t_a/t_b\right)_{des} = 2$$

$$t_b = 40F$$

$$\Omega_{des}/\Omega = 0.98 \quad \text{for} \quad t_a < (t_a)_{des}$$

$$\Omega_{des}/\Omega = 1.02 \quad \text{for} \quad t_a > (t_a)_{des}$$

The second method for temperature regulation is one which employes a radiator by-pass to meet the varying heat load conditions and to maintain a constant flow rate to the equipment at a constant inlet temperature. This approach simplifies the pump and drive motor design and permits them to be operated at or near optimum speed. Another advantage of this arrangement is the constant fluid flow at constant inlet temperature to the equipment which permits closer temperature tolerance and at or near optimum temperature range for the equipment.

For the varying heat load condition, the use of the radiator by-pass requires a corresponding variation in the radiator surface characteristic and/or the surface area, (ϵ A). There are several approaches which may be used to achieve the required changes in the product (ϵ A), which were mentioned in the previous quarterly report, Reference 1, pages 47-48.

The following equation was derived to show the relationship between the coolant outlet temperature from the radiator (t_{ro}/t_{bdes}) and the variation in the hat load (q/q_{des}) and by-pass flow rate (W_R/W_T) for the equilibrium case.

$$\frac{t_{r_o}}{t_{b_{des}}} = 1 - \frac{q_E}{q_{E_{des}}}\left(\frac{t_a}{t_b}-1\right)_{des}\left(\frac{W_T}{W_R}-1\right)$$

Figure 59 is a graphical presentation of the above equation. To achieve the required coolant outlet temperature from the radiator, the corresponding variation of the radiator area and/or emissivity (ϵ A) may be determined from the equation:

$$\frac{\epsilon A}{(\epsilon A)_{des}} = \frac{\Omega_{des}}{\Omega}\,\frac{q_E}{q_{E_{des}}}\,\frac{\left[t_{b_{des}}\left(\frac{t_a}{t_b}\right)_{des} + 460\right]^4}{\left[\frac{q_E}{q_{E_{des}}}\left(\frac{t_a}{t_b}-1\right)_{des} t_{b_{des}} + \left(t_{b_{des}} + 460\right)\right]^4}$$

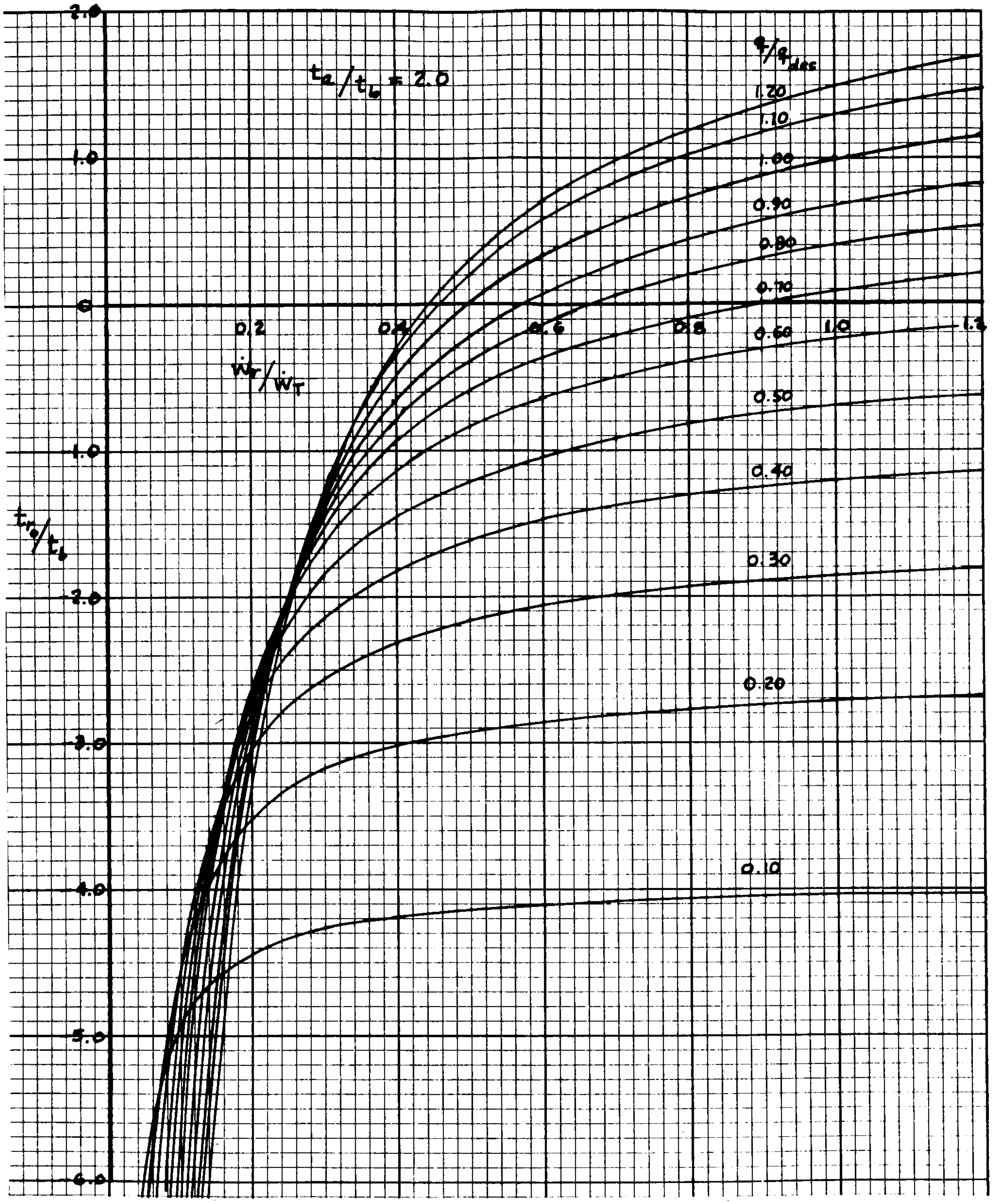


Figure 58. Radiator Outlet Temperature Versus Flow Rate, No Radiator By-pass

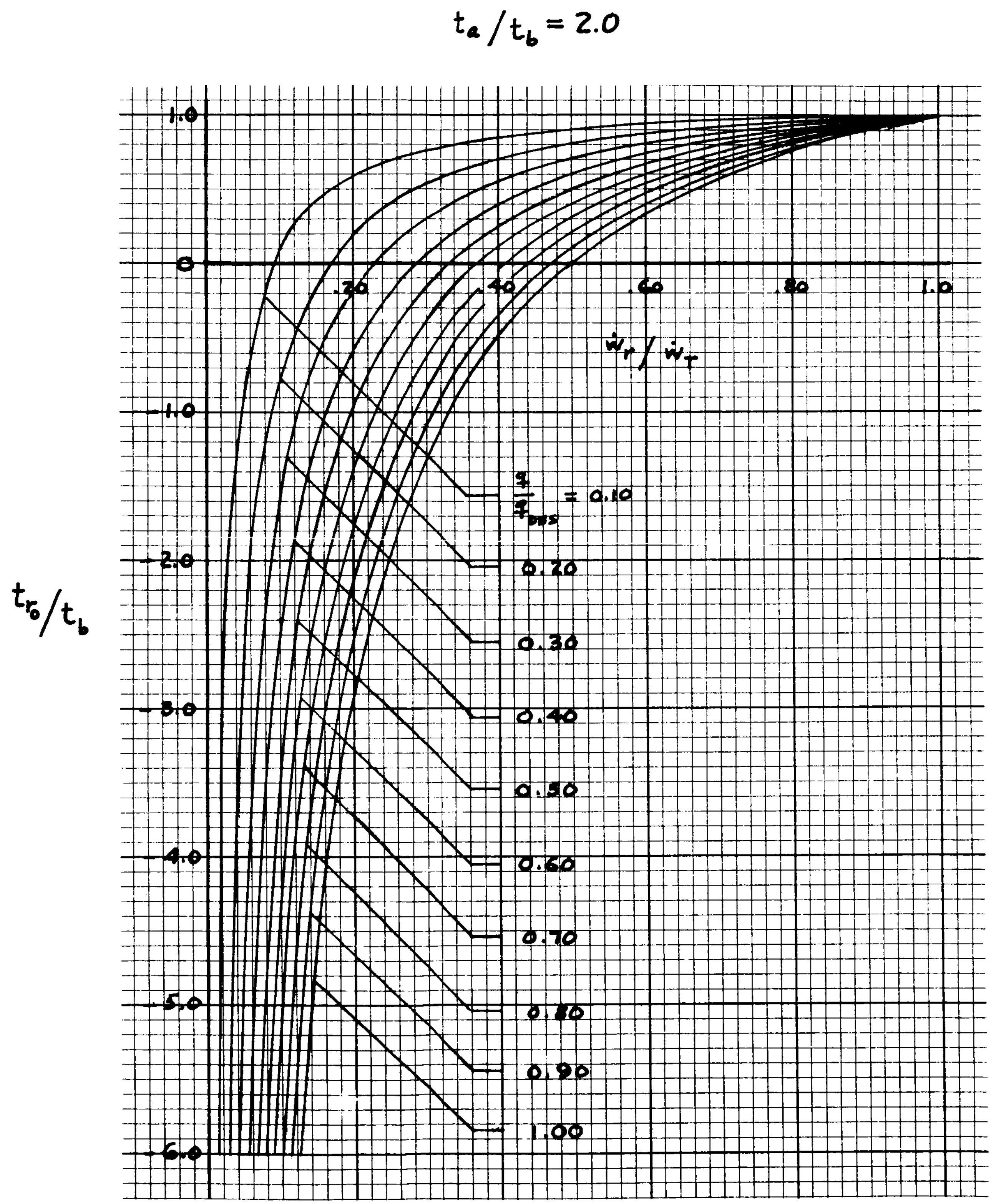


Figure 59. Radiator Outlet Temperature Versus Flow Rate, Radiator By-pass

Figure 60 is a graphical presenation of the above equation for two values of the radiator effectiveness ratio.

In Figures 59 and 60, the following values were assumed:

$$(t_a/t_b)_{des} = 2 \quad \text{and} \quad (t_b)_{des} = 40 \text{ F}$$

2.3.6 Heat Sinks

Most of the energy used to operate electronic equipment ultimately appears as heat in the immediate environment of the equipment. The heat is removed by either active or passive method or both and is disposed in a sink. There are a number of heat sinks that may be used with the various heat removal methods (active or passive). The selection of the heat sink or sinks depends upon the amount of heat to be rejected and the duration for heat rejection which will result in a minimum weight system.

The various heat sinks and the regions of applicability are illustrated in Figure 61. The regions or boundaries have been established on the basis of providing a minimum or least weight heat sink. In order to develop these regions of applicability, a number of assumptions and estimated values have been used. For this reason, Figure 61 represents an initial attempt to provide a very useful means for selecting heat sink to meet a particular requirement. Additional refinements in Figure 61 will be made and possibly several heat sink maps will be made for different assumptions and values for the numerous parameters or variables involved. The following is a brief discussion on the various heat sinks.

Thermal Mass

All equipment has some mass associated with it. The heat which appears when the electrical energy is dissipated causes the equipment mass to raise in temperature. This is directly proportional to the energy involved and inversely proportional to the heat capacitance (mass times specific heat). When the amount of heat is very small (in relation to the heat capacitance), the temperature rise is also small and no further thermal consideration is necessary. As the duration and heat loads increase, more energy must be absorbed. Additional mass can be used but this will result in considerable weight penalty. If a 20 F rise in temperature is permitted, even the best material will absorb only about 20 Btu/lb of material. For example, aluminum will absorb only about 4 Btu/lb and steel, only about 2 Btu/lb. The thermal mass is applicable for extremely small power dissipations or extremely short power spikes or where the material serves some other useful function. This type of heat sink can be regenerated (temperature reduced to original value) indefini-

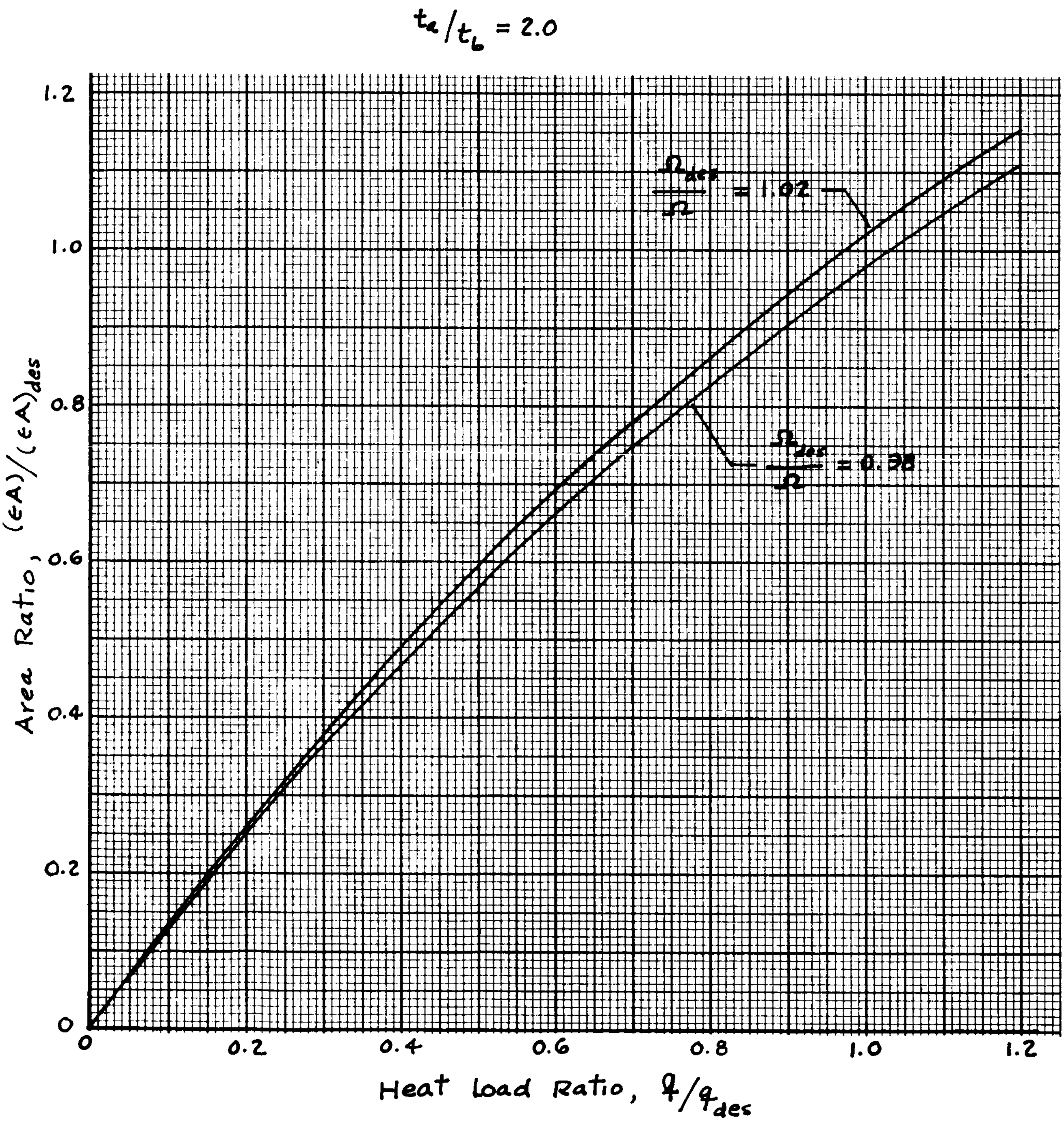


Figure 60. Radiator Area Ratio Versus Heat Load Ratio

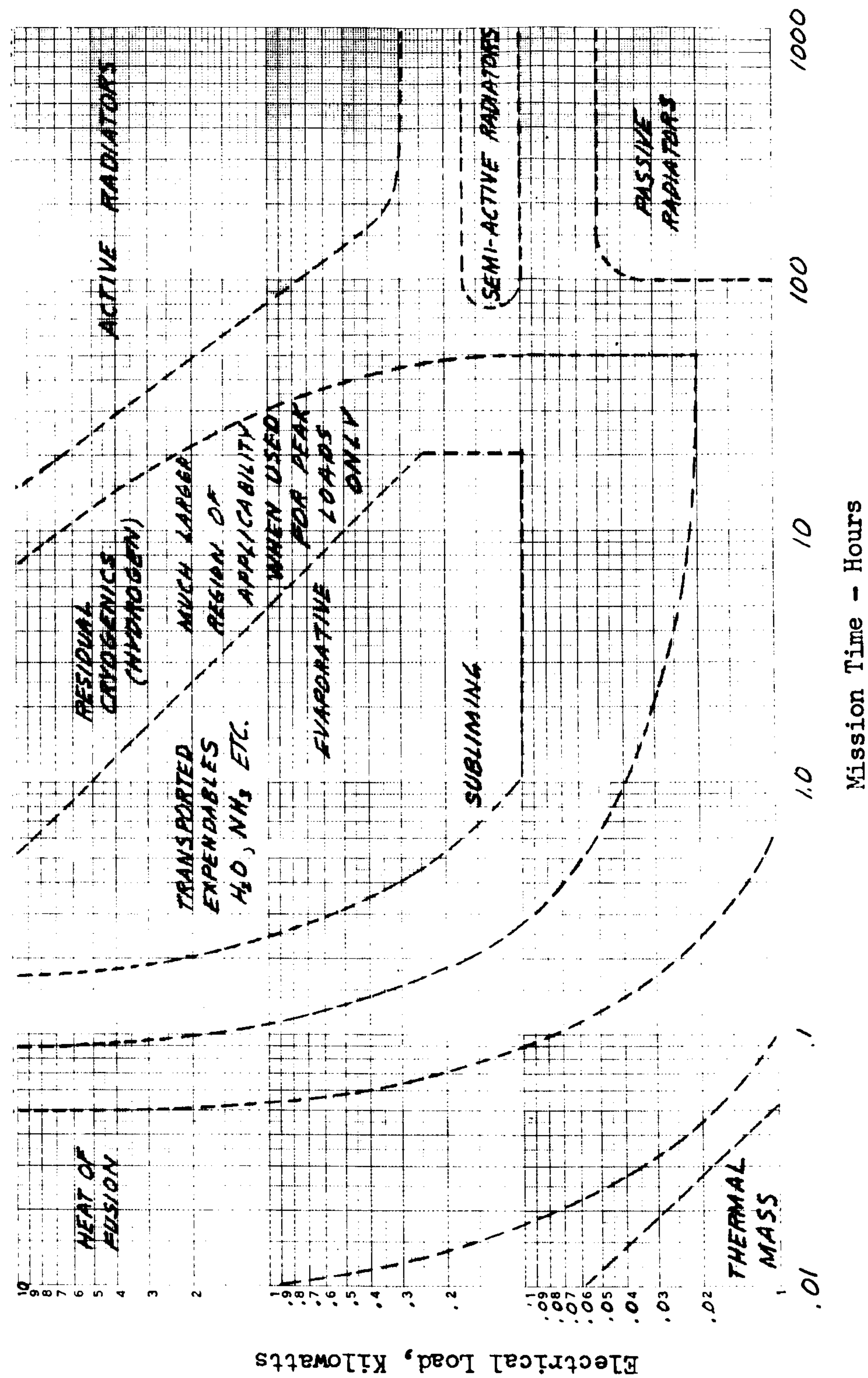


Figure 61. Regions of Applicability for Heat Sinks

tely, thus the useful capacitance may be several times the actual heat capacitance.

Heat of Fusion

When the total heat to be dissipated becomes appreciably larger, a more efficient heat sink is necessary. A number of materials which undergoes a change in state from a solid to a liquid may be used to absorb heat. As much as 138 Btu/lb can be removed in this manner. The heat of fusion limits the temperature rise of the electronic equipment to a narrow temperature band. Heat is also absorbed below and above the change of state temperature by capacitance. The containment of the change of state material must be flexible enough to account for some change in volume. This heat sink can be regenerated so its useful heat capacitance may be several times the heat of fusion. Figure 62 shows the heat of fusion for several different substances at the temperature the change of state occurs. Additional information can be obtained from Reference 12.

Expendables

The use of expendables as a heat sink has been discussed in Section 2.2.2 and the utilization of residual hydrogen in Section 2.3.3. The information provided in these two sections were used to establish the regions of applicability in Figure 61.

The weight penalty for utilization of expendables include the lines, pressurization, coolers, connectors,valves, etc. necessary to utilize the expendables. In many instances the utilization of available expendables will not be practical. However, the ability of expendables to be used on a demand basis makes the use of expendables particularly attractive for peak (spike) loads since they reduce the size of radiators and make their load more uniform.

Radiation

The theory and application of radiating surfaces for heat rejection to space is well known and is widely used in past and current space vehicle thermal system. Since space can be considered to be an infinite heat sink, the radiating surface or radiator is perhaps the most efficient method for long duration, low to high heat loads. The radiator may be classified as passive, semi-active and active, depending upon the design.

A passive radiator is one in which the equipment to be cooled is attached directly to the radiating surface. This is a simple arrangement with no moving parts or power requirements. Since no fluids are

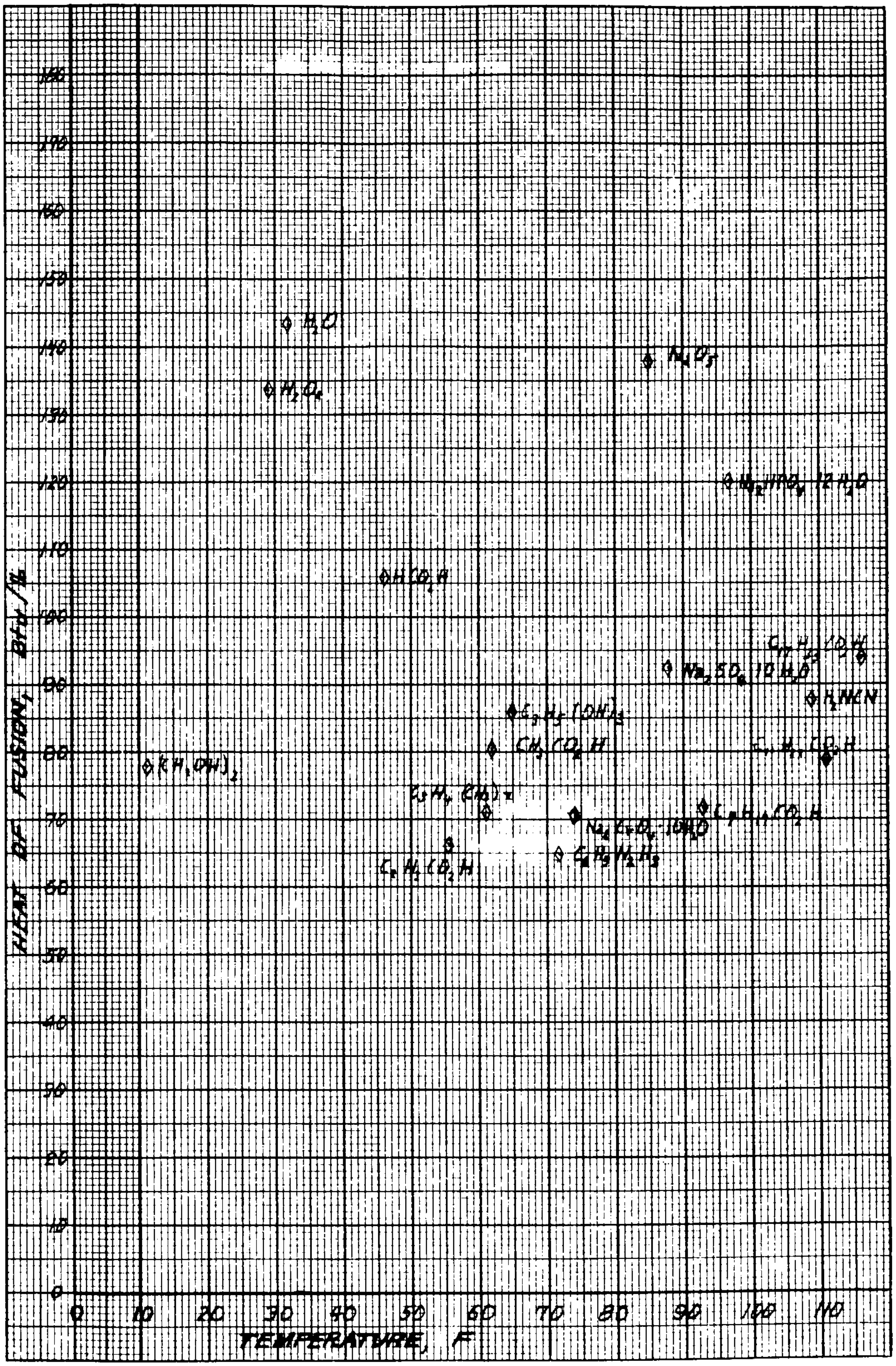


Figure 62. Heat of Fusion Data

involved, there are no problems of freezing or meteoroid puncture. It is limited in its application to constant heat loads or to equipment with large temperature tolerance.

A radiator in which a heat pipe with fluid flow but no external power or moving parts is used to transfer heat from the equipment to the radiating surface is classified as semi-active. This method provides some self-regulation to accommodate variations in the heat rejection load and/or environmental heat load. A closer temperature tolerance may be maintained by the use of the heat pipe as compared to the passive method. Relatively large passages for the heat pipe are required which pose a problem of meteoroid penetration if large areas or long duration use is required.

A radiator is classified as an active radiator when a circulating fluid is used to transfer heat from the equipment or source to the radiating surface. This method provides the maximum temperature regulation to accommodate variations in heat loads. The use of fluids creates the problems of freezing and possible radiator tube punctures by meteoroids. For long term application, the meteoroid protection must be considered, and will have some influence on the radiator weight.

2.4 INTEGRATED SYSTEM SYNTHESIS AND ANALYSIS

The selection of the optimum thermal control concepts for the various astrionic equipment, must be based on an integrated thermal system which is a combination of the electronic package environmental control method and the vehicle thermal system. This requires the synthesis and analysis at various integrated systems to establish the preferred ones. Among the important consideration for the selection of the preferred systems are: heat load capability, weight, power requirements, volume, area, reliability, integration potential, and availability. These factors are considered in the synthesis and analysis of the integrated systems.

In the following paragraphs, various system concepts are presented which are considered to be preliminary and these and additional concepts will be investigated in the next quarter. This is followed by a discussion on the system reliability.

2.4.1 Thermal System Concepts

A number of thermal control system concepts are shown schematically in Figures 63 and 71. These are described below with regard to some general characteristics, and no attempt has been made in the schematics to optimize the number and type of control components (valves, sensors, etc.). This work will be continued in the next quarter when more of the details will be defined.

In Figures 63 through 71, the following symbols are used to designate the various components of the system:

CP = cold plate
P = pump
PR = pressurization source
SR = space radiator
C = compressor
HX = heat exchanger
GSE = ground support equipment

The concept illustrated in Figure 63 consists of a liquid coolant loop which picks up the electronic packages heat load in a cold plate circuit and rejects this load to an expendable

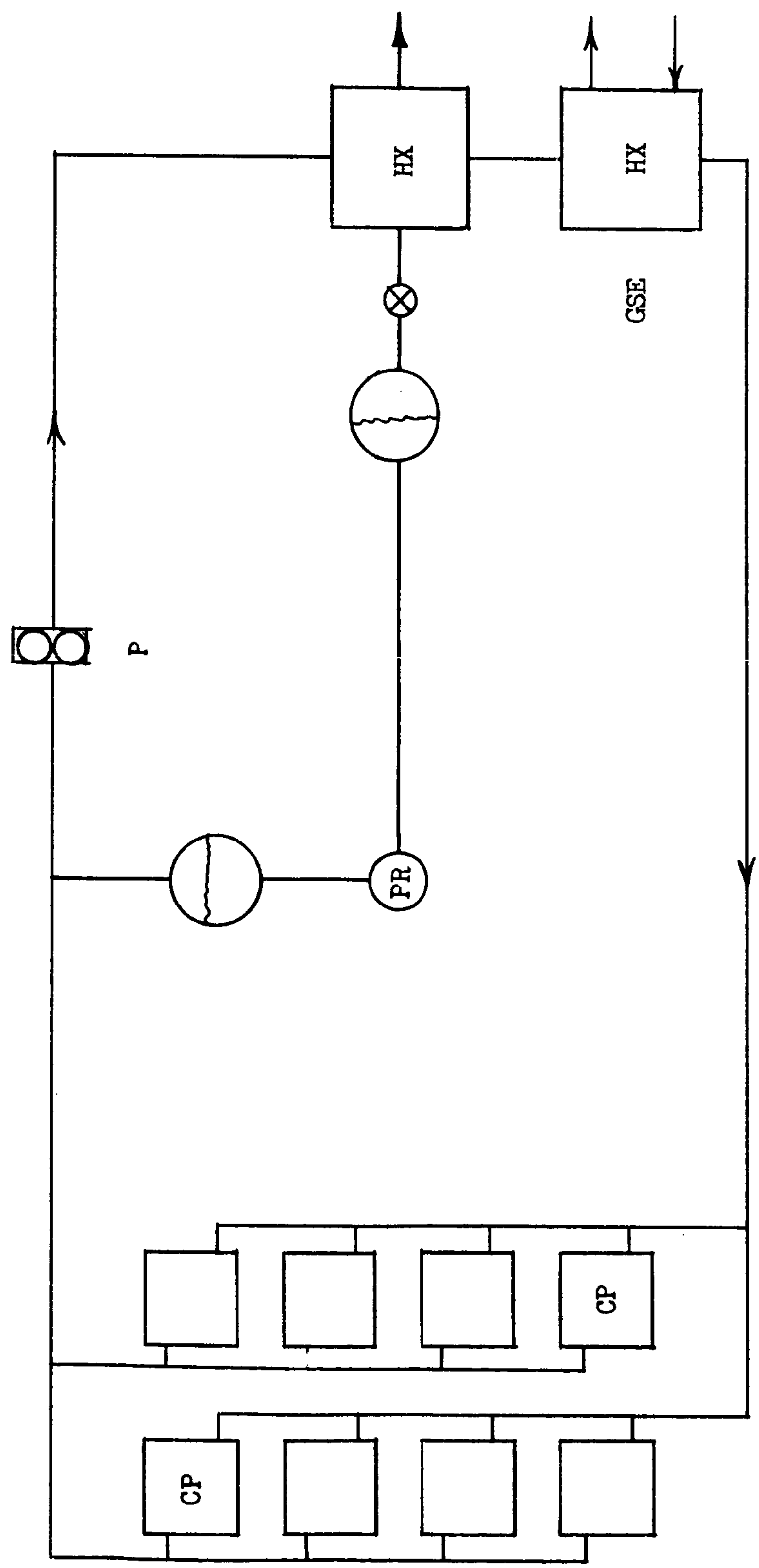


Figure 63. Expendable Heat Sink System Concept

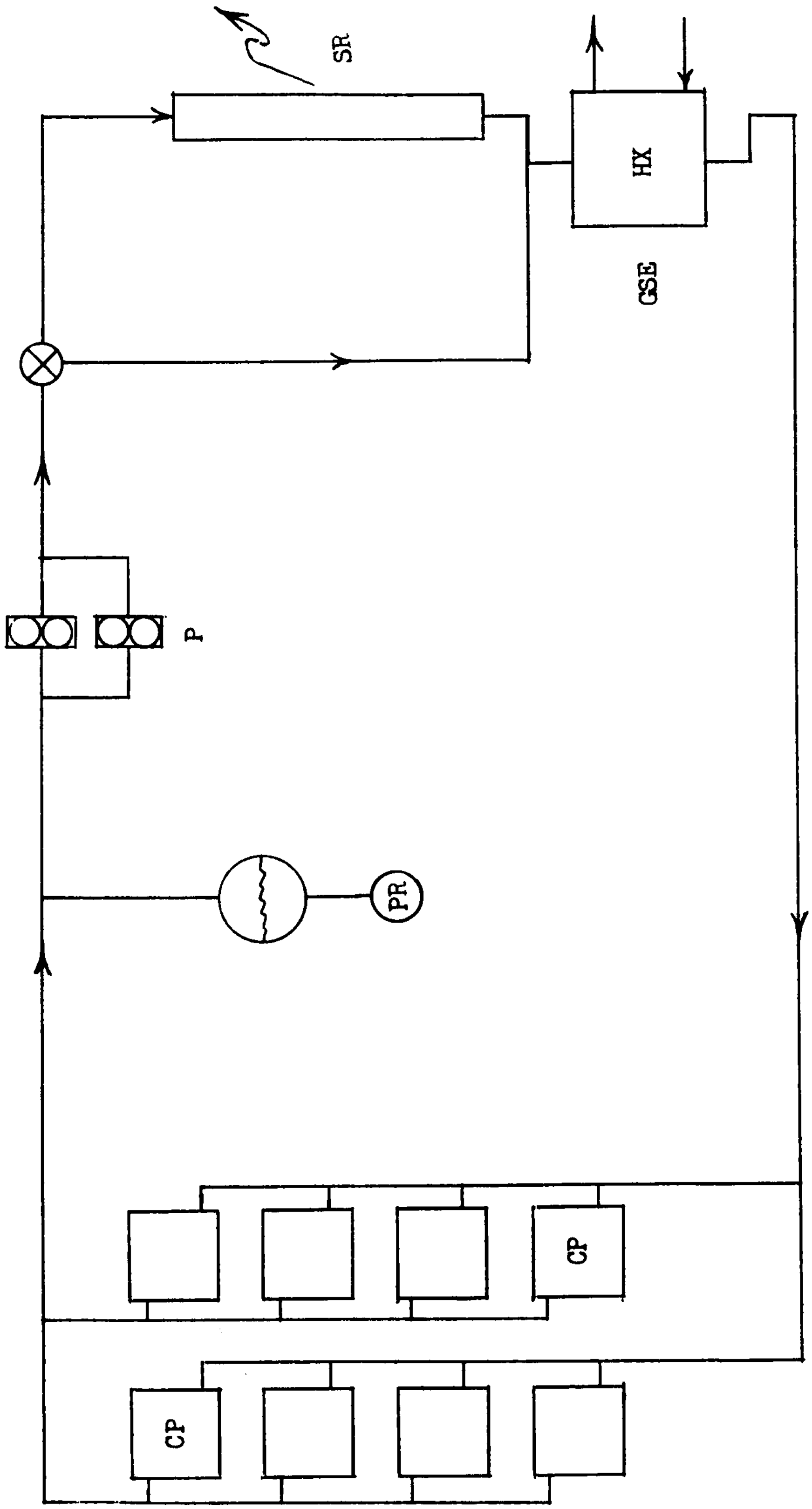


Figure 64. Closed-Recycle Coolant Loop Concept

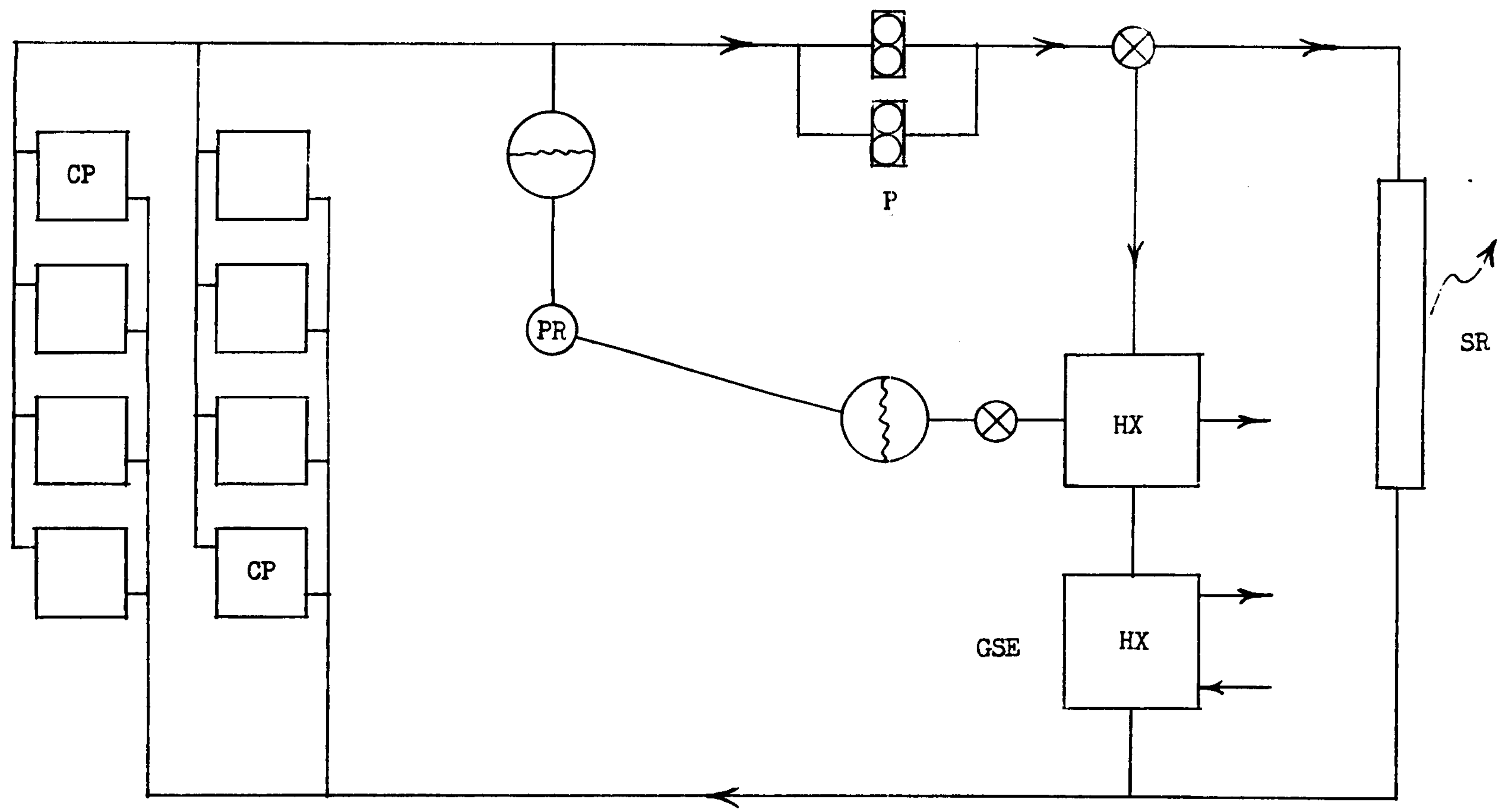


Figure 65. Closed-Recycle Coolant Loop and Expendable Heat Sink Combination Concept

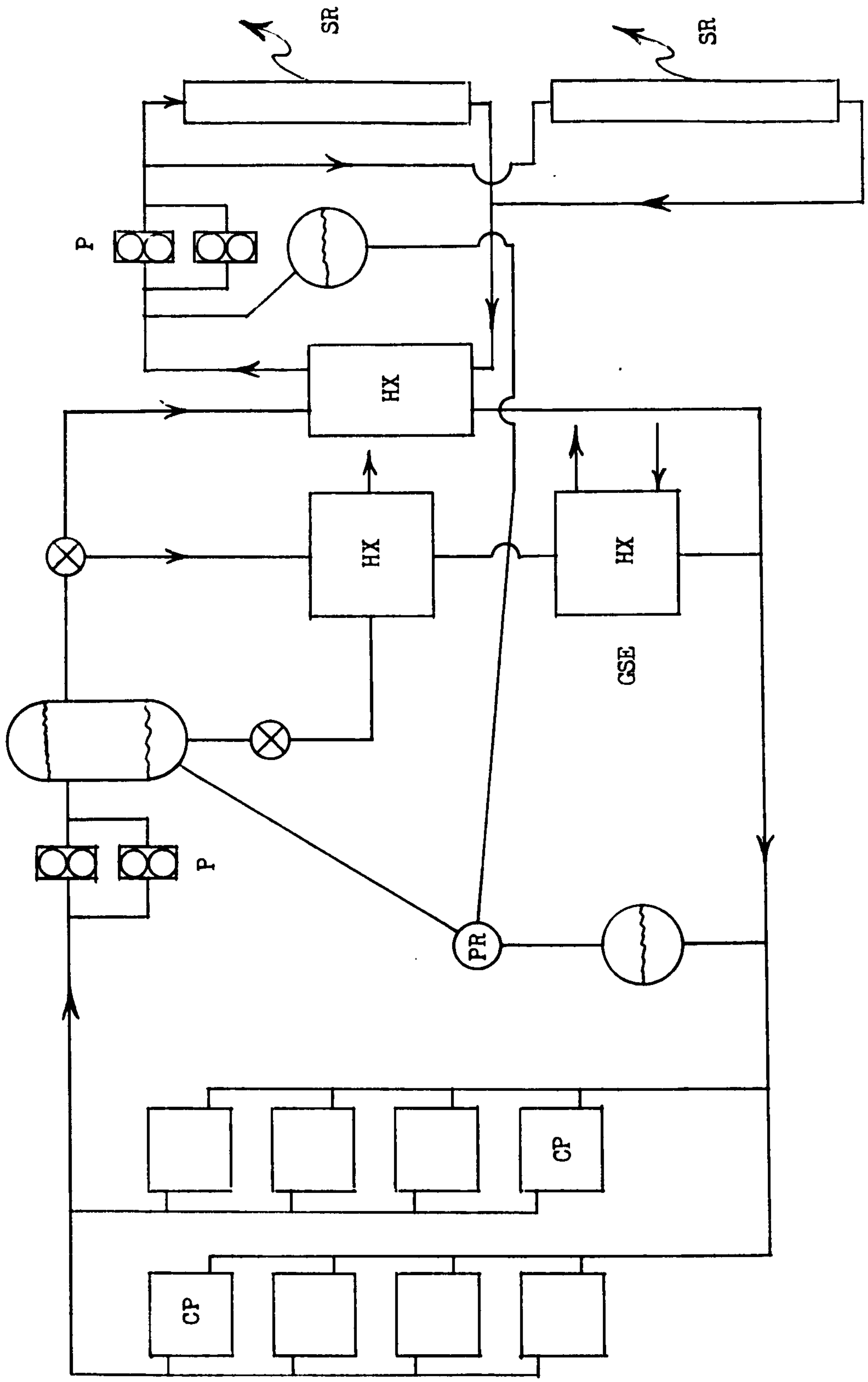


Figure 66. Dual Closed-Recycle Coolant Loops with Expendable Heat Sink Combination Concept

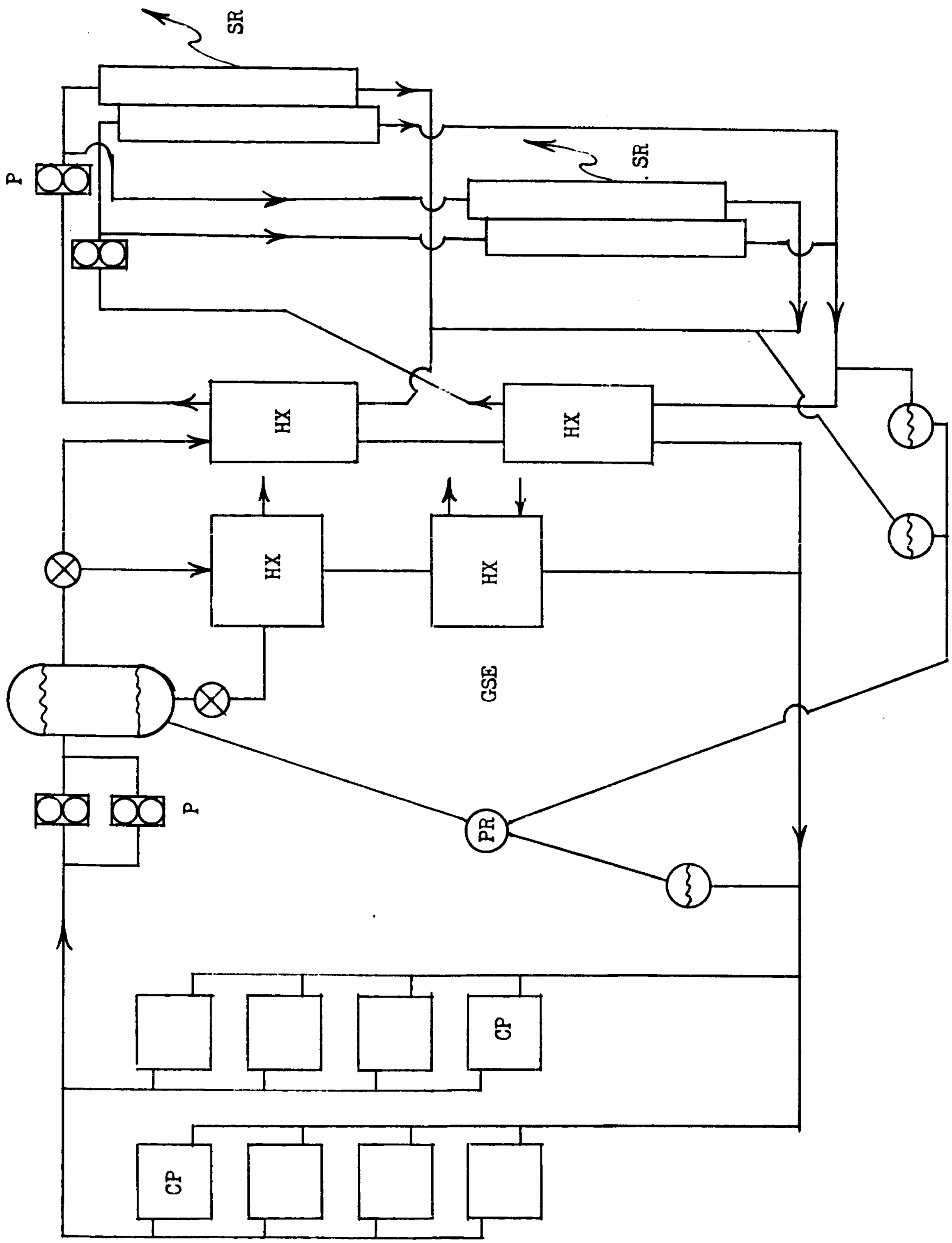


Figure 67. Dual Closed-Recycle Coolant Loops with Redundant Radiator Liquid Loop Concept

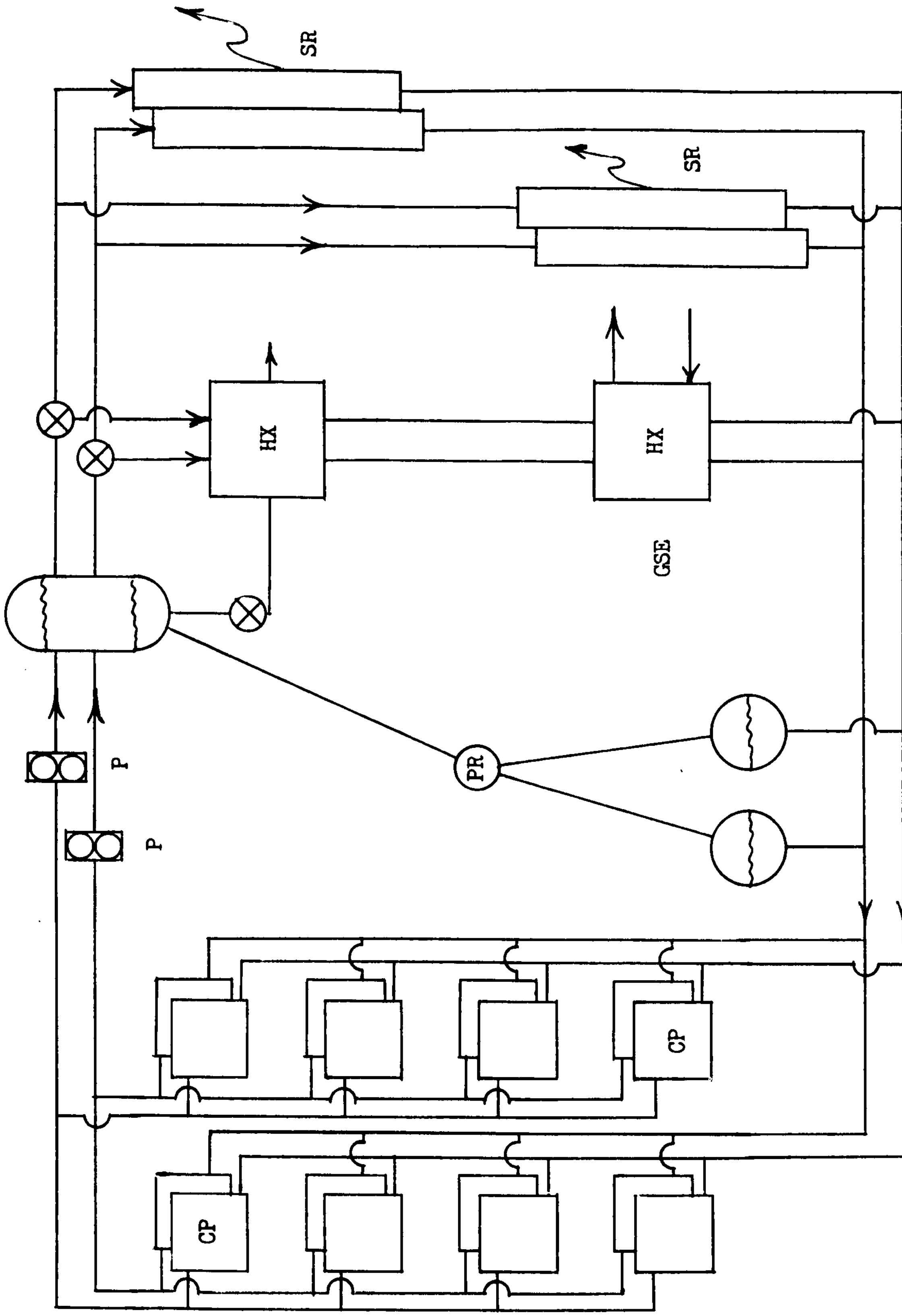


Figure 68. Redundant Closed-Recycle Coolant Loops and Expendable Heat Sink Combination Concept

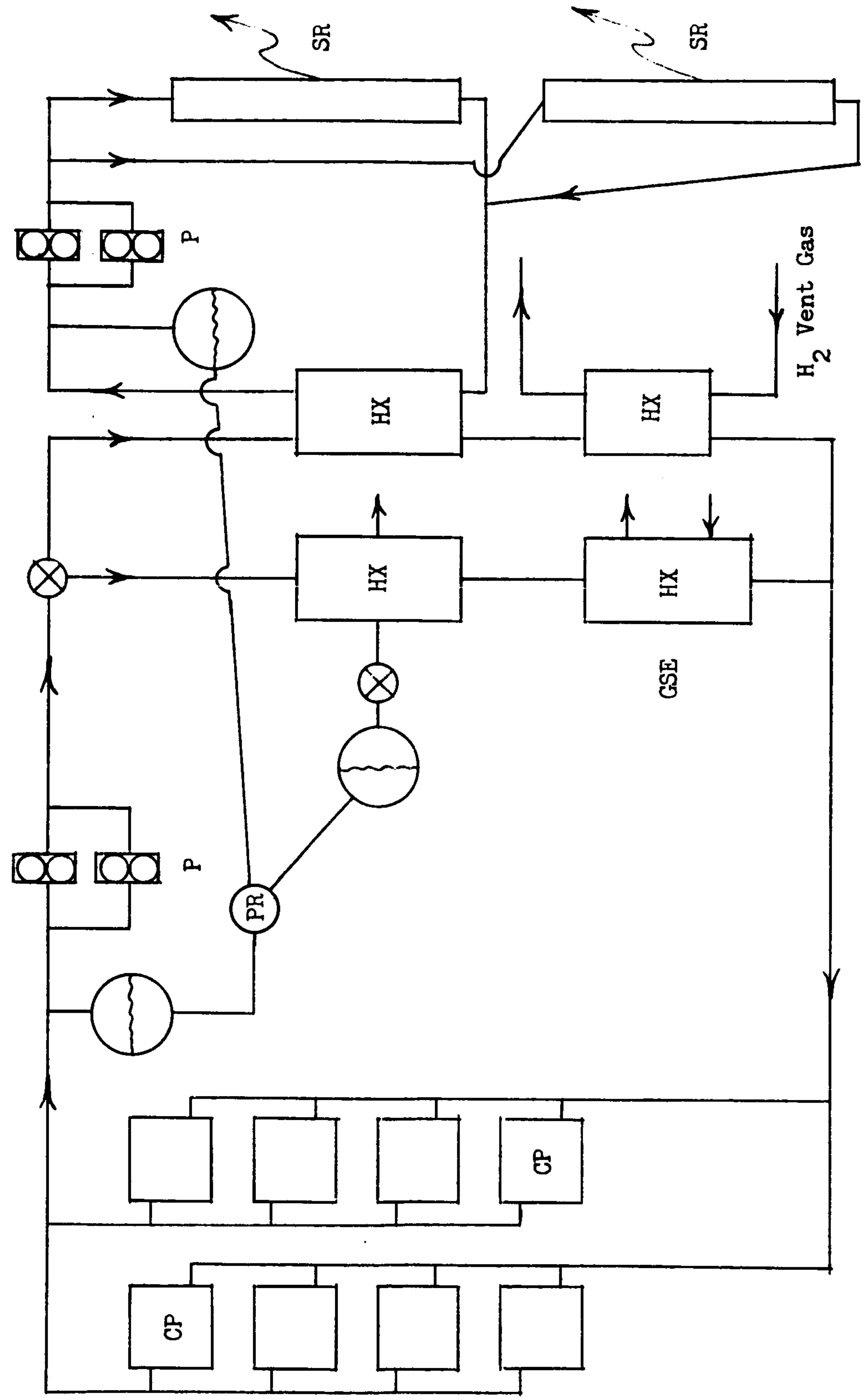


Figure 69. Dual Closed-Recycle Coolant Loop with Vent Gas Utilization Concept

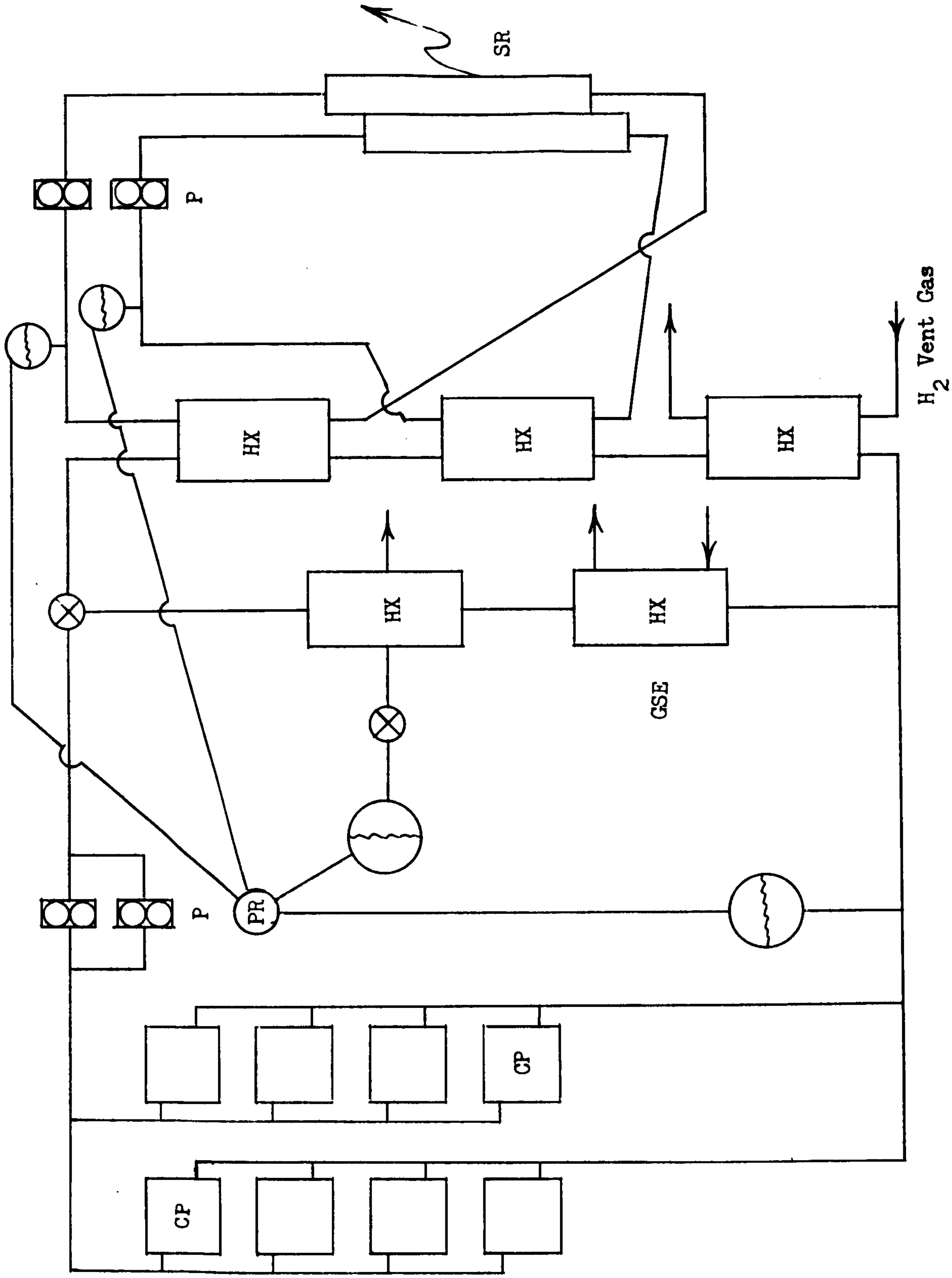


Figure 70. Dual Closed-Recycle Coolant Loop with Redundant Radiator Liquid Loop and Vent Gas Utilization Concept

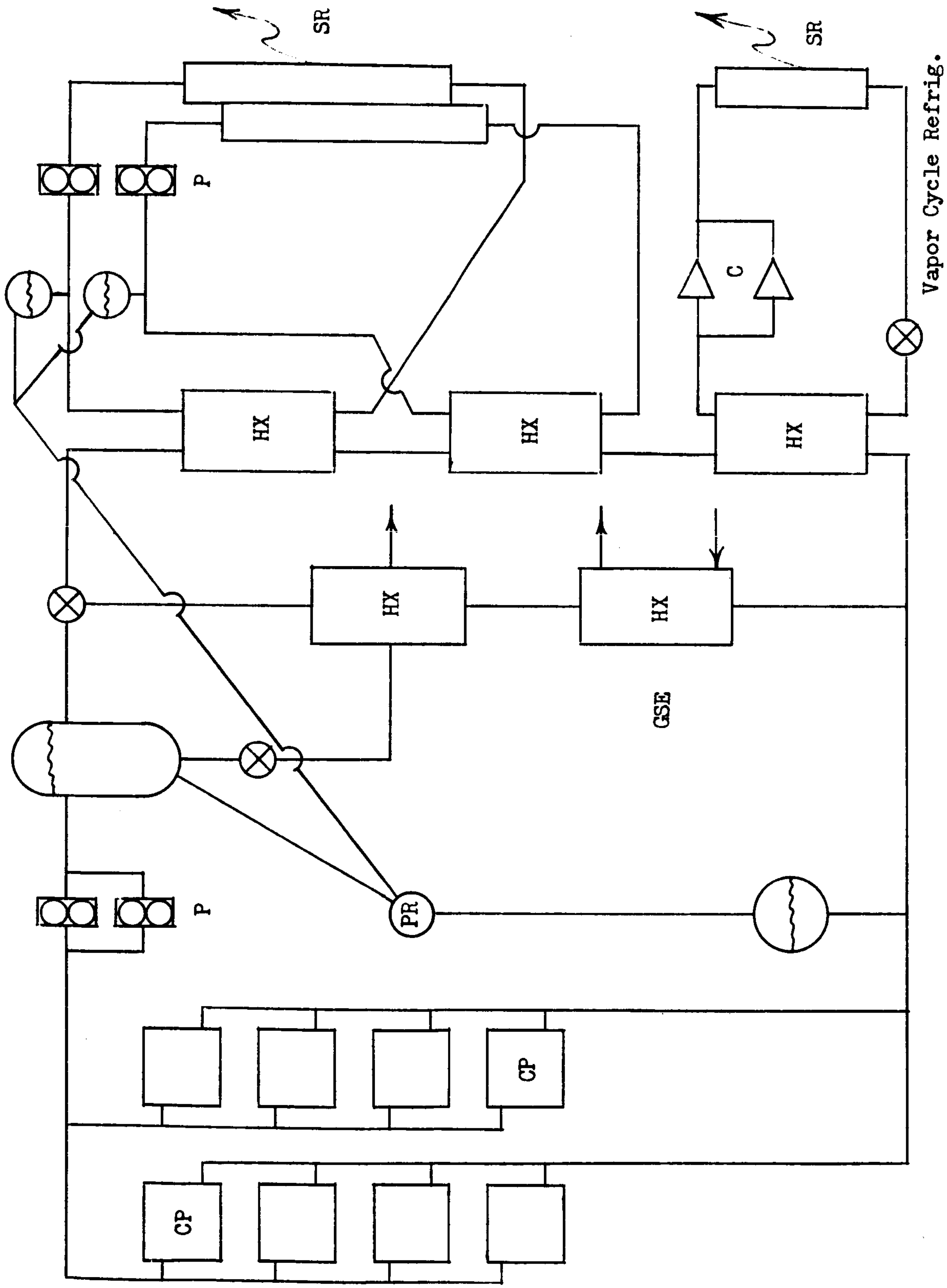


Figure 71. Dual Closed-Recycle Coolant Loop with Vapor Cycle Concept

heat sink (such as water). This system has the advantage of having few components subject to failure, but system weight becomes prohibitive for mission durations in excess of a few hours due to the increasing weight of expendable which must be carried. The system contains a coolant circulating pump and a water boiler as well as a liquid-to-liquid heat exchanger. The latter is connected to the launch pad conditioning system and is used only until lift-off. A water-storage tank and a coolant accumulator are also shown in the schematics. The accumulator serves to maintain the pressure at the inlet to the coolant pump at the desired level; and a source of high-pressure nitrogen gas is used for both liquid loop pressurization, and expulsion of water from the storage tank. Temperature control of the coolant is obtained through regulation of the rate of evaporation in the boiler, which must have sufficient capacity to handle the maximum load expected during a particular mission. Also, since mission duration is limited, it may not be necessary to provide for standby equipment to take the place of components which have failed. Hence, no duplicate components are shown in the Figure 63 schematic.

To overcome the short-duration limitation of an expendable heat sink system, a closed-loop liquid coolant system can be employed. In such a system, shown in Figure 64, the electronic packages heat load is transported by the coolant to a space radiator which serves to reject the heat load to the deep space heat sink. The radiator must be sized such that its capacity is sufficient for the maximum expected heat load and under the most severe ambient conditions likely to be encountered. Temperature control of the coolant is obtained by means of a bypass around the space radiator; and coolant loop pressure is maintained through an accumulator pressurized by a source of nitrogen gas. As in the expendable sink system, a liquid-to-liquid heat exchanger has been included to provide for cooling prior to lift-off. (This type of heat exchanger is shown in all system schematics and serves the same purpose in all systems.) Because the closed-loop liquid system is applicable to extended duration missions, dual coolant pumps are included in the schematic. Since the pump is considered to be the component most likely to fail under constant running conditions, a standby unit provides some measure of protection against a complete shutdown of the cooling system.

A variation of the closed-loop coolant system is shown in Figure 65. In this schematic, the bypass around the space radiator includes a water boiler which can be used to augment the space radiator under peak heat load conditions. This arrangement permits the radiator to be designed for the maximum steady-state condition and avoids the excess capacity requirement of the basic schematic in Figure 64. Dual coolant pumps are provided for redundancy, and nitrogen gas is again used for accumulator pressurization and expulsion of water from its storage tank.

Further refinement of the liquid coolant loop technique is illustrated in Figure 66, which represents a two-loop arrangement. One coolant loop transports the electronic packages heat load to a liquid-to-liquid heat exchanger, where the heat load is transferred to the coolant in the second loop. The latter serves to transport the heat load from the heat exchanger to a space radiator for rejection to the deep space heat sink. The advantage of this two-loop system is that a complete failure of the outer loop, the one including the space radiator, does not cause an immediate shutdown of the cooling system. The inner loop can function independenly of the outer loop by making use of an emergency expendable heat sink, which is indicated in Figure 66. This expendable heat sink can also serve to absorb a portion of peak heat loads under normal operating conditions. The warm coolant leaving the electronic packages cold plates is shown to pass through the water storage tank to keep the water in the tank from freezing. This precaution may be necessary for long duration missions if the water storage tank is subject to heat loss. Another feature of the concept shown in Figure 66 is the dual space radiator provision. In the event that one of the radiators suffers damage as a result of meteoroid impact, it is isolated from the system and the other radiator takes over the function of the damaged one. The two radiators may be combined in a single panel which contains two separate fluid passages between fin surface areas. It should also be noted that the liquids in the two coolant loops need not be the same because there is no physical connection between the two fluid streams.

The cooling system concept illustrated in Figure 67 is similar to the one shown in Figure 66 except that it includes two outer coolant loops which are completely independent. The advantage of this arrangement is that, in the event of excessive coolant loss from one loop, another is available to take its place. Although both this and the previous schematic show the accumulator for the inner loop upstream of the cold plates, it is pointed out that this arrangement was done purely for convenience. In an actual system, the accumulator will generally be near the pump inlet because it is the point of lowest system pressure.

A different means of providing redundancy is shown in Figure 68. The concept illustrated is similar to the one in Figure 65 except that two completely independent coolant loops are provided. With this design, no single component failure can induce a complete cooling system shutdown. Of course, the penalty for carrying the extra weight of the dual system must be evaluated in terms of the alternatives, which include the possibility of a premature shutdown of the cooling system and mission failure.

Primary cooling system performance can also be augmented through utilization of the sensible heat capacity of cryogenic hydrogen vent gas, which may be available during certain missions. Utilization is implemented through a suitable liquid-to-gas, counterflow heat exchanger, as illustrated in Figure 69. The addition of this heat exchanger is the only basic difference between the schematic shown here and the one in Figure 66.

Another variation of a cooling system which utilizes hydrogen vent gas is shown in Figure 70. This concept is similar to the previous one, except for addition of a second, outer coolant loop which is completely independent. As indicated earlier, the provision of this degree of redundancy must be evaluated in terms of applicable penalty factors.

The last of the conceptual arrangements considered up to this point is shown in Figure 71. In this case, the heat rejection capacity of the liquid coolant system under peak heat load conditions is supplemented by a vapor cycle refrigeration system. Aside from this feature, the concept shown is similar to the one in Figure 67. The heat exchanger in the refrigerant loop is essentially an evaporator which absorbs heat from the liquid coolant, and the space radiator acts as a condenser. Because rotating components are considered to be subject to failure, a dual compressor system is indicated.

All of the foregoing concepts were devised on the basis of providing some degree of alternate heat rejection capacity in case of system or component failure. Possible major failures may occur in:

(1) moving parts, such as pumps or compressors
(2) radiators, through puncture or other damage causing leakage
(3) heat exchangers, through leakage, and
(4) valves and/or control systems

A secondary failure may be considered to be leakage at joints or connections which develops after some period of system operation.

2.4.2 System Reliability

General Considerations

Prime aspects in the attainment of highly reliable heat transport fluid systems for electronic package thermal conditioning are examined. Design considerations are discussed with respect to averting or minimizing the general types of failures to be anticipated in liquid systems applicable to missions up to 180 days duration.

The basic system is considered to consist of a closed recirculating loop containing manifolded "cold plates", space radiators for heat rejection, circulating pump, and liquid accumulator(s) to maintain pressurization and accommodate differential expansion coefficients of the metal system and contained liquid. The basic system will not be complete without the addition of: GSE shielding, preconditioning and checkout disconnects, a thermal regulation system of sensors, sensor conditioners, program logic controller, power amplifiers, power driven valves, and usually provision for liquid filtration. Additional items that may be provided include heaters to prevent fluid congealing in radiators, heat exchangers for peak load management, or interfacing with other available environmental control systems to support or receive support from such systems.

The basic system may be required to operate continuously, or intermittently, remain dormant for a major portion of its life or perform according to on-board preprogrammed or remotely supplied commands. In addition, the system must accommodate the higher aero-thermo launch and perhaps re-entry regimes plus vehicle orientation changes and permit operation during prelaunch checkout on the pad and during associate contractor equipment integration and assembly at the launch site. Interconnection of the system thermal loop while in space transit or orbit with the ECS of other vehicles that may rendezvous also may be desired. During ground operation there may be the requirement of avoiding problems associated with collection of water condensed from the atmosphere.

Ideal operating conditions for a recirculating liquid loop would be those approaching a continuous steady state which would permit a minimum of equipments and a control consisting of the switch to apply power to the circulating pump motor. When redundancy of motors and coolant loops are provided the minimum of monitoring will increase to permit system condition evaluation and transfer of functions to the alternate equipment. As the system function becomes further removed from steady state to meet operating mode changes and thermal transfer variations the equipment items including sensors and controls plus redundancy for contingencies will increase further requiring more detailed attention per item to assure the same level of reliable operation.

Reliability Aspects of the Liquid Loop. Principal reliability design objectives generally common to liquid loop systems are the provision for assured containment and pressurization of the liquid volume and the maintenance of its circulation. Thermal transfer capability will be impaired or lost whenever the integrity of any one of these three objectives is degraded. Progressive loss of the thermal transfer fluid from the loop leads to ultimate system failure. Internal loop leakage causing bypassing or short circuit conditions also can affect system functions significantly. Inadequate liquid pressurization will permit cavitation of the pump, insufficient circulation, and probable pump failure. The foregoing failure modes are associated with system leakage (with possible exception of a spring loaded rather than gas pressurized accumulator) that result in thermal transport failures. Circulation deficiencies also can fail the system by causes which include corrosion product or foreign matter plugging, pump, valve or bypass control system failure and radiator or other system element failure and radiator or other system element freezing.

Design Approaches for Reliability. The three principal reliability objectives (liquid containment, pressurization and circulation) provide general criteria for good design, approaches when the potential failure modes discussed above are considered.

Ideal System. An ideal liquid containment approach suggests the use of a completely contained, hermetically sealed unit requiring only a clamping in place and the attachment of signal, control and power leads. Any integration of the system with other equipment would avoid fluid disconnects for installation, checkout, servicing and interfacing with GSE. Practical considerations require compromise with the ideal approach; however the fluid interfaces may be minimized and backup design measures plus effective closeout procedures will provide effective results approaching the ideal system.

Leakage Contingency. Further assurance of system success is available by use of two completely separate loops so that possible leakage of one will not result in loss of fluid in the other. Two channels in the individual cold plates and space radiator panels plus dual lines and auxiliary equipment are necessary. Additional backup cooling may be obtained by use of hydrogen vent gas when available. Thermal transfer may be achieved by a liquid gas heat exchanger or possibly by direct admission of the gas to appropriate liquid channels after the liquid is purged from necessary portions of the system. The later method would require in addition to sizing and control considerations, the precaution to avoid blockage by freezing if the liquid is dumped into space.

Pressurization. Liquid pressurization may be achieved by gas pressurization with a bladder liquid interface contained in a tank-sized to provide reserve fluid for minor leakage loss and perhaps a heat sink afforded through prelaunch liquid chilling and valving. Liquid pressurization may be provided by means of spring loaded bellows type accumulator located at the suction side of the recirculating pump(s). The sizing of the bellows type accumulator generally provides a rather limited reserve of liquid useful to make-up fluid loss. Redundancy of the bellows system in a given loop does not appear warranted for leakage or spring failure. The accumulator liquid reserve usually is relatively small and spring failure is considered to have a low probability of occurrence. The gas pressurized accumulator is dependent upon the integrity of the bladder and the support of a regulated gas pressurization system. The Apollo command module utilizes both types of accumulators in a primary loop and the smaller bellows accumulator only in the backup loop.

Circulation. Assurance of recirculating flow can be enhanced by the use of standby redundancy of the pump unit so that failure of either unit does not cause system loss. Sensing for evaluation and switching controls will be necessary adjuncts to this approach. Pump design and selection will emphasize zero external leakage and long life for adequate levels of reliability under continuous or intermittent duty.

Possible circulation stoppage resulting from corrosion or breakdown products plugging orifices, filters or valves can be minimized by selection of materials for compatibility, adequate filter sizing, and equipments least sensitive to fouling. Overpressure relief or bypass and other "fail safe" approaches are additional measures to assure circulation continuity.

<u>Control.</u> The application of standby redundancy for high levels of reliability requires the use of sensing equipment for system condition evaluation by vehicles or ground crew/onboard computer, and the necessary command and control for function transfer. General practice in the manned Apollo system is to provide standby redundancy of some sensors in each loop plus redundancy in the signal conditioning and control amplifier systems. Utilization of the redundant items in some portions of the system is exercised by manual switching of electrical connectors in the crew compartment to place the alternate or standby unit in operation. Unmanned systems would provide this function with additional complexity which may be warranted upon special evaluation of the particular functions and their relation to the mission objectives. Transfer or isolation of some system functions by manual valve override also is provided in the manned Apollo systems.

Other controls which may be tied to the basic thermal loop controls are those necessary for effective operation of equipment such as an evaporative heat exchanger. System control of the evaporative fluid admission and its rate of evaporation by steam valve outlet pressure regulation will be required. The steam outlet valve may require heating to prevent freezing and consequent impaired efficiency in use of the expendable evaporative liquid. Fail safe or override control may be considered in the detail design approach to a specific system application.

A thermal control system which is to remain "dormat" for a significant portion of the mission will require analysis of start up requirements and the capability of satisfactorily providing adequte reliability. A frozen space radiator would require defrosting by raising temperature above the congealing condition that the circulating pump could handle. Lower freezing point liquids or continuous heat load or reorientatoon of the appropriate space radiator panels are methods for considerations in the intermittent or dormant system. Thermal system controls should also be capable of start up operation from the dormant state upon receipt of appropriate commands. This portion of the system may require a controlled environment to maintain the ready state condition.

System Effectiveness Analysis

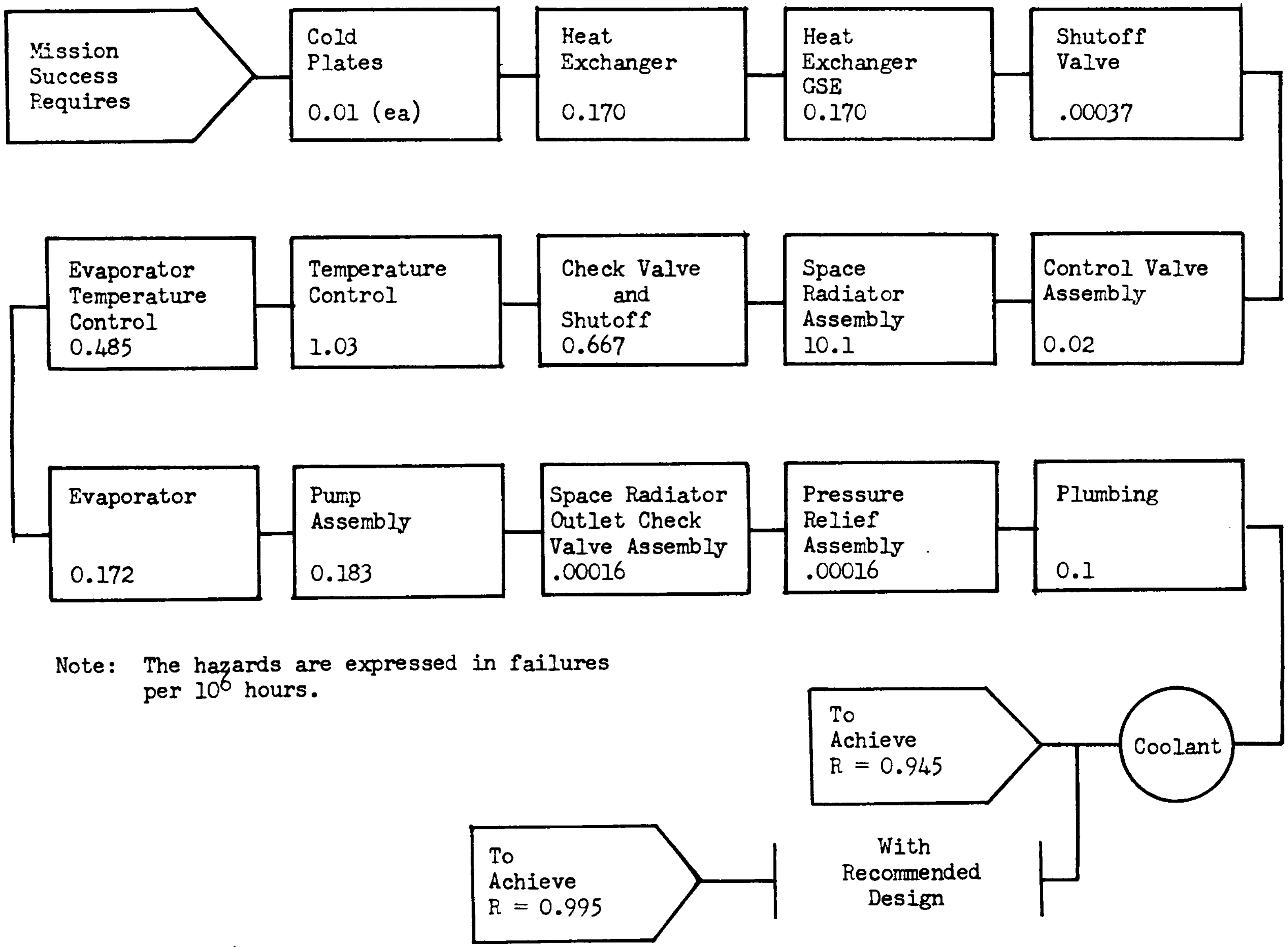
The reliability and system effectiveness becomes very closely related for the unmanned system because the man-machine interface is ineffective after launch. Such is the case for the thermal system under study. To faciltate a study of the reliability/effectiveness problem, a simplified logic diagram was prepared and is given in Figure 72. The components described by each block are at the assembly level, those typical of the Apollo type design. For each block a failure hazard was estimated, based on Apollo II level but projected into the 1975 time frame.

S&ID data indicates that the reliability of any state of art system when expressed in terms of failure hazard, can be improved by a factor of between 5 and 10 for any given five year period. This assumes that a reasonable effort is maintained to accomplish this improvement. Using these data, the estimated probability of mission success for the "single thread" or non-redundant system would be no more than 0.945 for the 180 day mission. This in itself may be sufficient but when considered in conjunction with all the other missions may have to be "weekest link" and not meet its apportioned unreliability. A more desirable value was estimated to be greater than 0.995.

Given that the "single thread" design is unsatisfactory, a cursory weak link analysis was made as reflected in Table 16. The intent was to determine those elements that were introducing the higher failure hazards, determine the cause and/or failure mode and then determine a potential design corrective action. The results are reflected in the table, the objective can be achieved with little difficulty and with the contemporary technological capability.

Corrective Action Considerations. The major contributors to system unreliability and ineffectiveness are the space radiators and the cold plates. Their major failure modes are quite similar though some causations are different; they can either leak, clog or burst, in that order of relativity. These will be considered separately.

Space Radiators - can fail due to a meteoroid puncture, the probability of occurrence is inversly proportional to the wall thickness as shown in Figure 73. Recent data from Pagasus seems to indicate that the puncture hazard is very remote indeed, at least for Apollo type design. The dotted line on the figure reflects the estimated Apollo value and the recommended thickness. The burst failure mode is even more remote since reasonable design practice and proof testing can virtually eliminate the possibility. Clogging is a very real possibility as previously discussed, even after normal precautions.



Note: The hazards are expressed in failures per 10^6 hours.

Figure 72. Logic for a Simplified Thermal System

Table 16. Thermal System Weak Link Analysis and Suggested Design Sections

Functional Assemblies	Estimated Failure Hazard ($\times 10^6$)	Failure Hazard Goal ($\times 10^6$)	Required Design Action
1. Space Radiators	10.1	0.1	Separate into multi-section with separate flow control, use 6 to 8 separate loops.
2. Temperature Control Assembly	1.03	0.1	Use redundant assembly with an error sensor, non-operating.
3. Check Valve and Shutoff Ass.	0.667	0.1	Use double redundant check valves. Others acceptable.
4. Evaporator Temp. Control	0.485	0.1	Use redundant controller with error sensor.
5. Pump Assembly	0.183	0.183	Acceptable as is.
6. Evaporator	0.172	0.172	Acceptable as is.
7. Heat Exchanger	0.170	0.170	Acceptable as is.
8. Heat Ex. GSE	0.170	0.170	Acceptable as is.
9. Plumbing	0.10	0.10	Acceptable as is.
10. Control Valve	0.02	0.02	Acceptable as is.
11. Coldplates	(0.01 ea)	0.1	Where more than 10 are used they should be separated into several loops with flow control.
12. Shutoff Valve	.0004	.0004	Acceptable as is.
13. SR. Check Valve	.0002	.0002	Acceptable as is.
14. Pressure Relief	.0002	.0002	Acceptable as is.
System	13.098	1.358	Design as a "sealed" unit.

$$\delta = 2.5 \times 10^{-4}\, Kv^{\,1/3}\, (A\tau)^{0.30}\, f(P_{(0)})$$

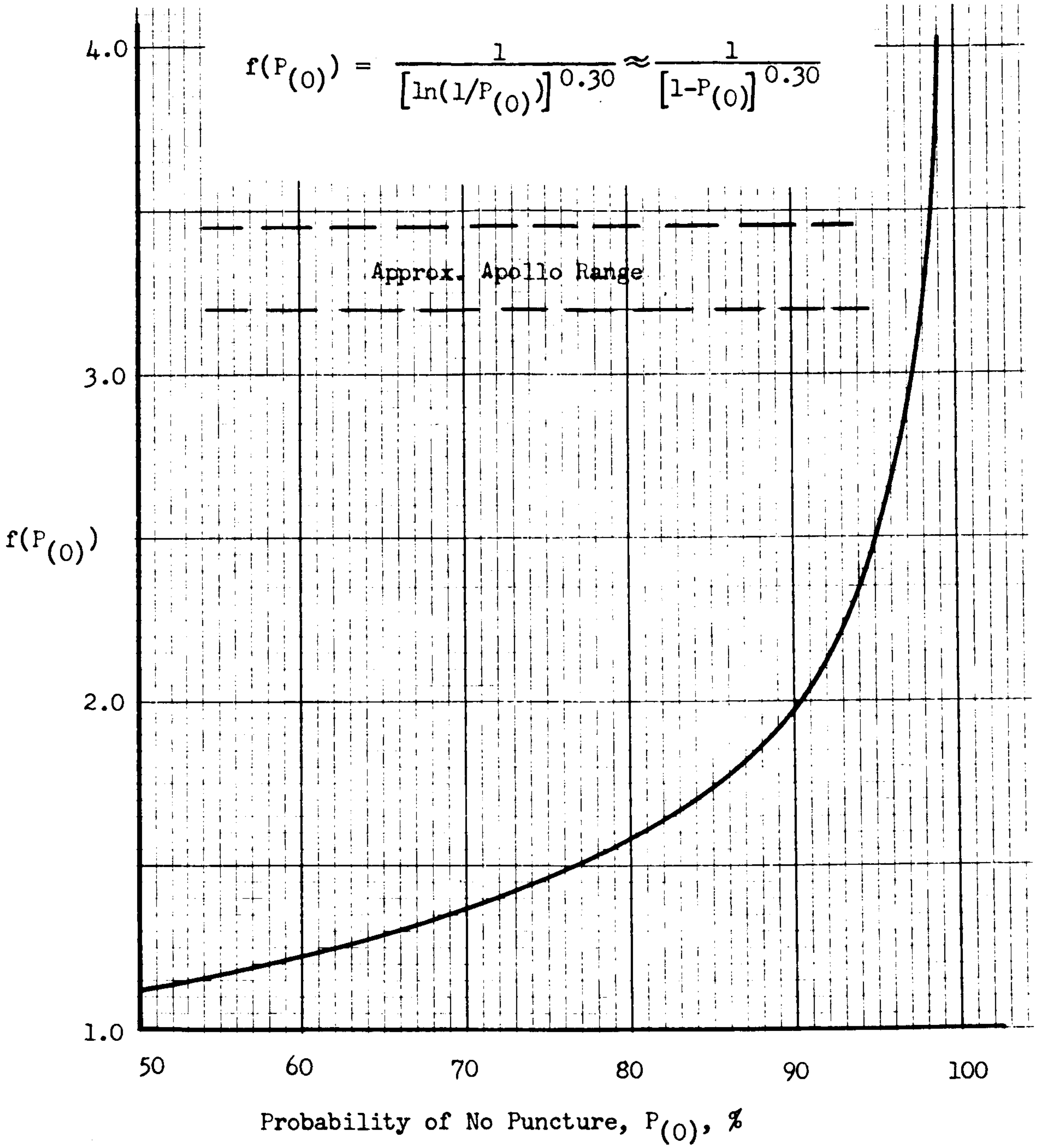


Figure 73. Probability of No Puncture Versus Thickness Factor

With all of the foregoing factors considered, the space radiators do present a high failure hazard. The best way of off setting this potential problem is illustrated in the recommended configuration and demonstrated by the curves of Figure 74. By dividing the total required area into a number of separable loops and slightly overdesigning for the total area, the space radiator functional reliability and effectiveness can be increased to meet any reasonable requirement. A minimum of four loops seems optimum in light of the complexity introduced by the larger number of loops.

The Cold Plates - present almost an equal level of unreliability, when all are considered a requirement. They also can plug up, leak or burst, in that order of expectancy. The arguments for and against are similar to the space radiators except that leaks will be caused by various forms of overstress. Plugging will be the most prominent failure mode and a parallel channel design within the plates combined with multiple loops for the main feed will reduce the hazard to an acceptable level as shown by Fiugre 75. The higher the number of loops, the less the chance of the total function (cold plate cooling) will be out of commission. The chance that one specific one will be out is acceptably low, but the chance that one of many will fail is sufficiently high to warrant dividing into several loops. It is recommended that no more than five plates be in any individual loop and where a concentration of heat requires a larger area, it should be separated or be partically cooled by two loops.

Preliminary Conclusions. - The available data indicates that a carefully designed system could probably achieve a reliability of 0.995 or better. A potential configuration, designed to achieve this goal and maintain a semblance of simplicity is presented in Figure 76. An approximation of the equivalent mission success logic is presented in Figure 77 where some of the multiple redundancy, possible with the recommended design, is demonstrated. The recommended design provides for up to six different operating modes, excluding cold plate considerations. Cold plate configuration alternatives are many and must be taylored to the very specific design; in any event, a minimum of four more alternatives are possible.

Increasing the reliability and subsequent effectiveness of a thermal control system does impose penalties in the form of additional weight and power consumption. Within reasonable goal limits, the penalties for high reliability are not expected to be excessive. The requirements are obviously a function of mission duration and/or duty cycle control. If all systems functions were in use (worse case) the effect on weight for high reliability would be as estimated in Figure 78. The 180 day mission with the suggested goal would raise the total weight by a factor of less than 1.5 over the non-redundant weight.

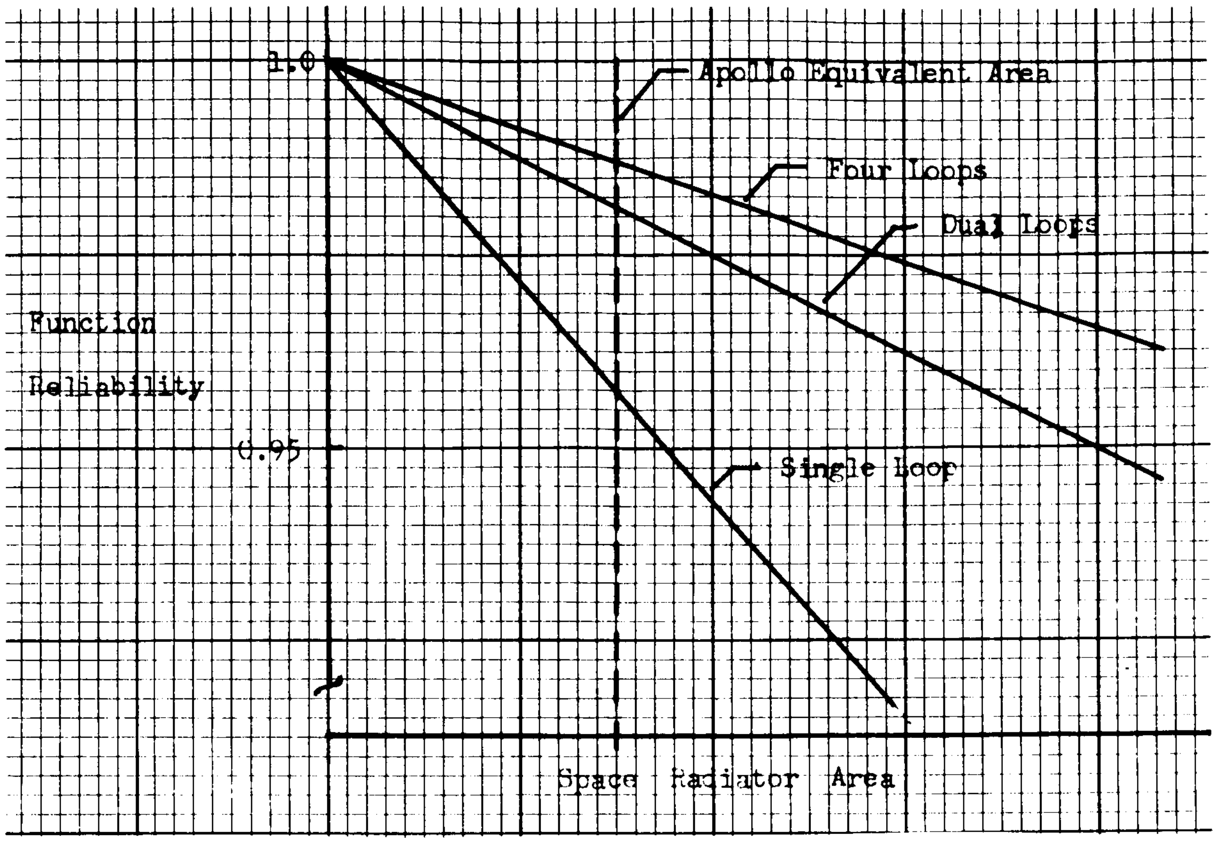


Figure 74. Reliability of a Space Radiator as a Function of Sectionalizing

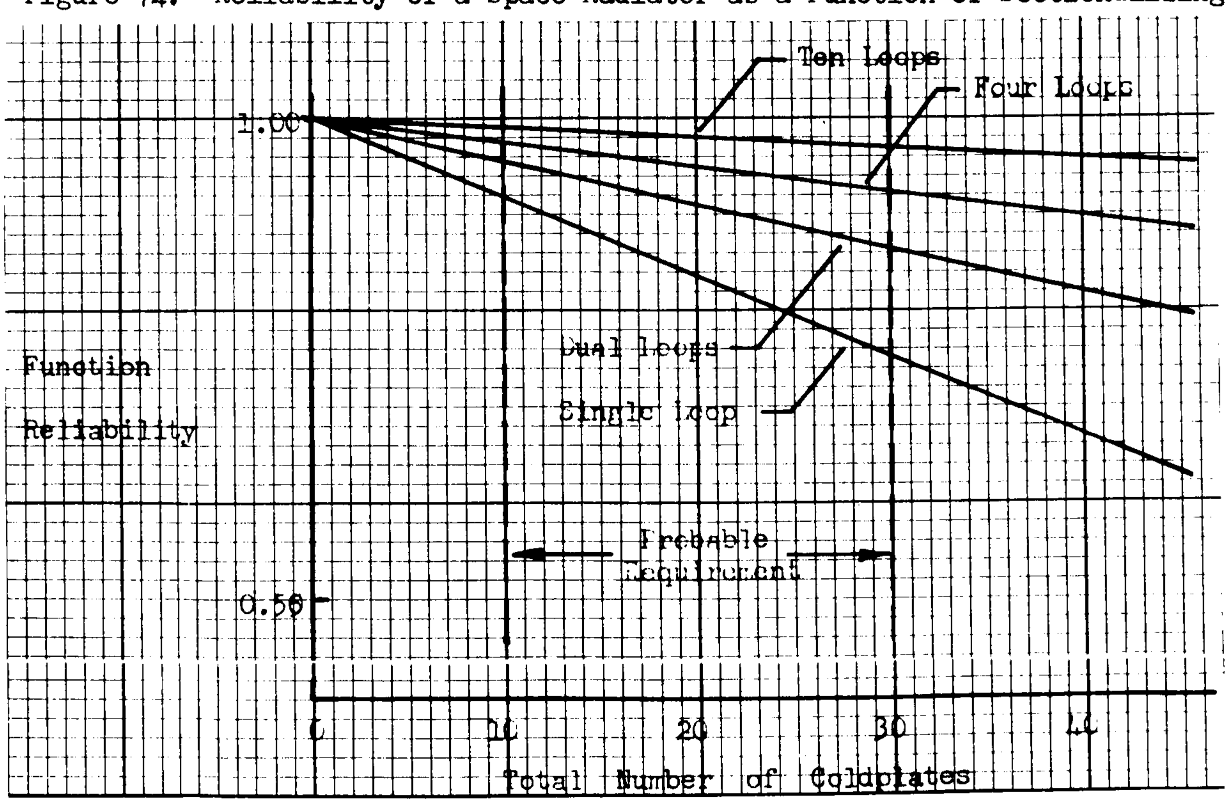


Figure 75. Coldplate Reliability Consideration

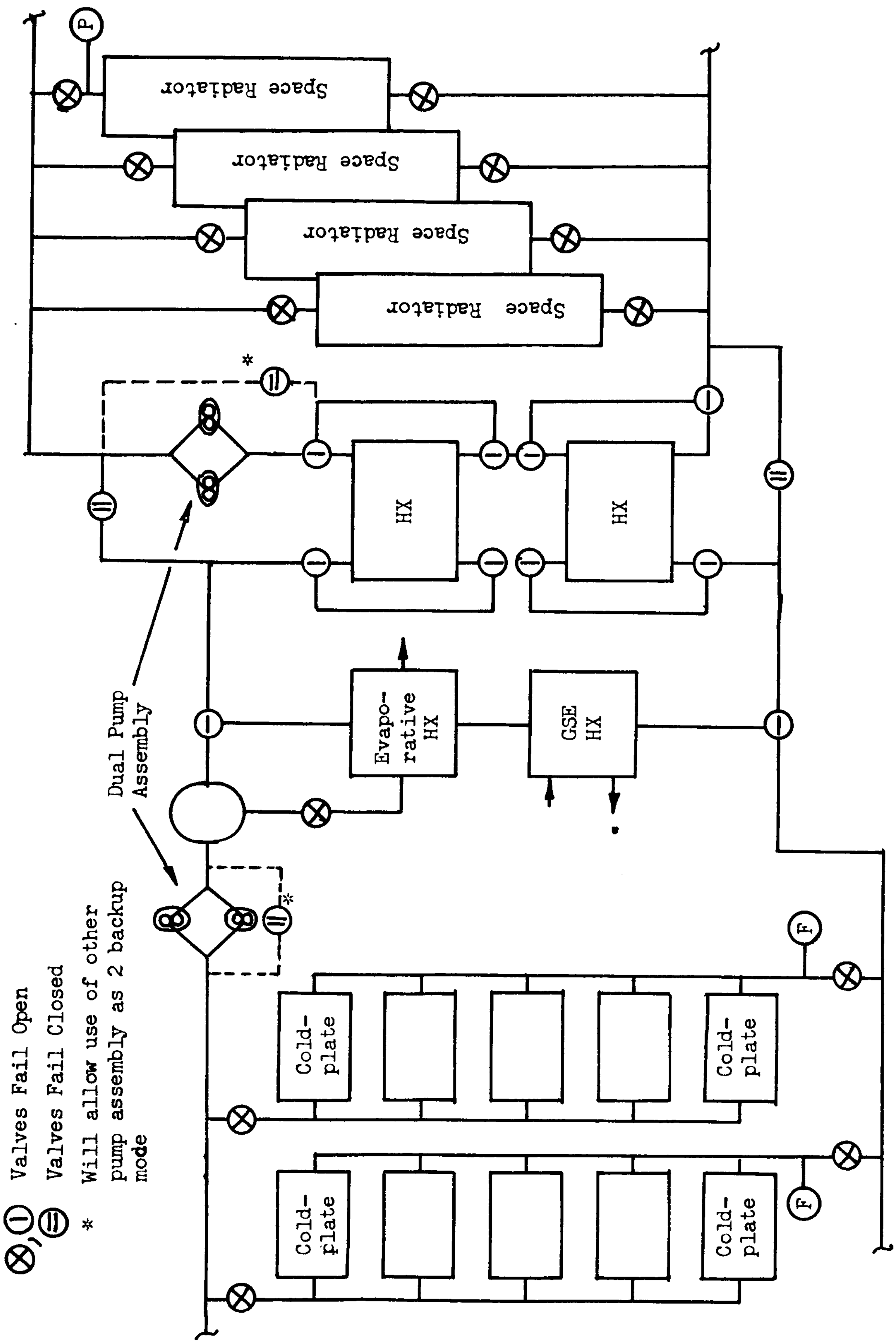


Figure 76. Potential Thermal Control System Configuration

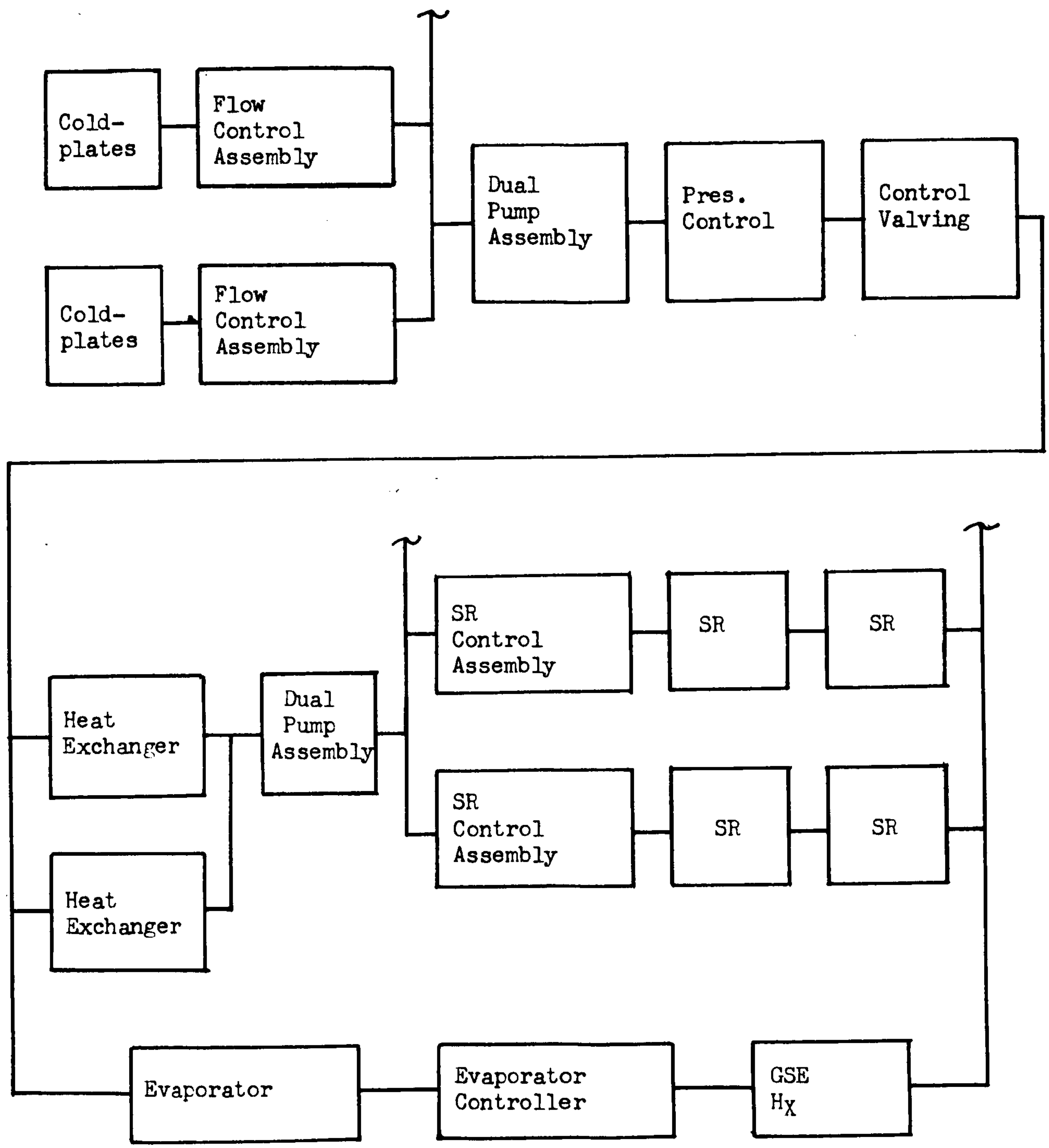


Figure 77. Revised Mission Success Logic For The Potential System

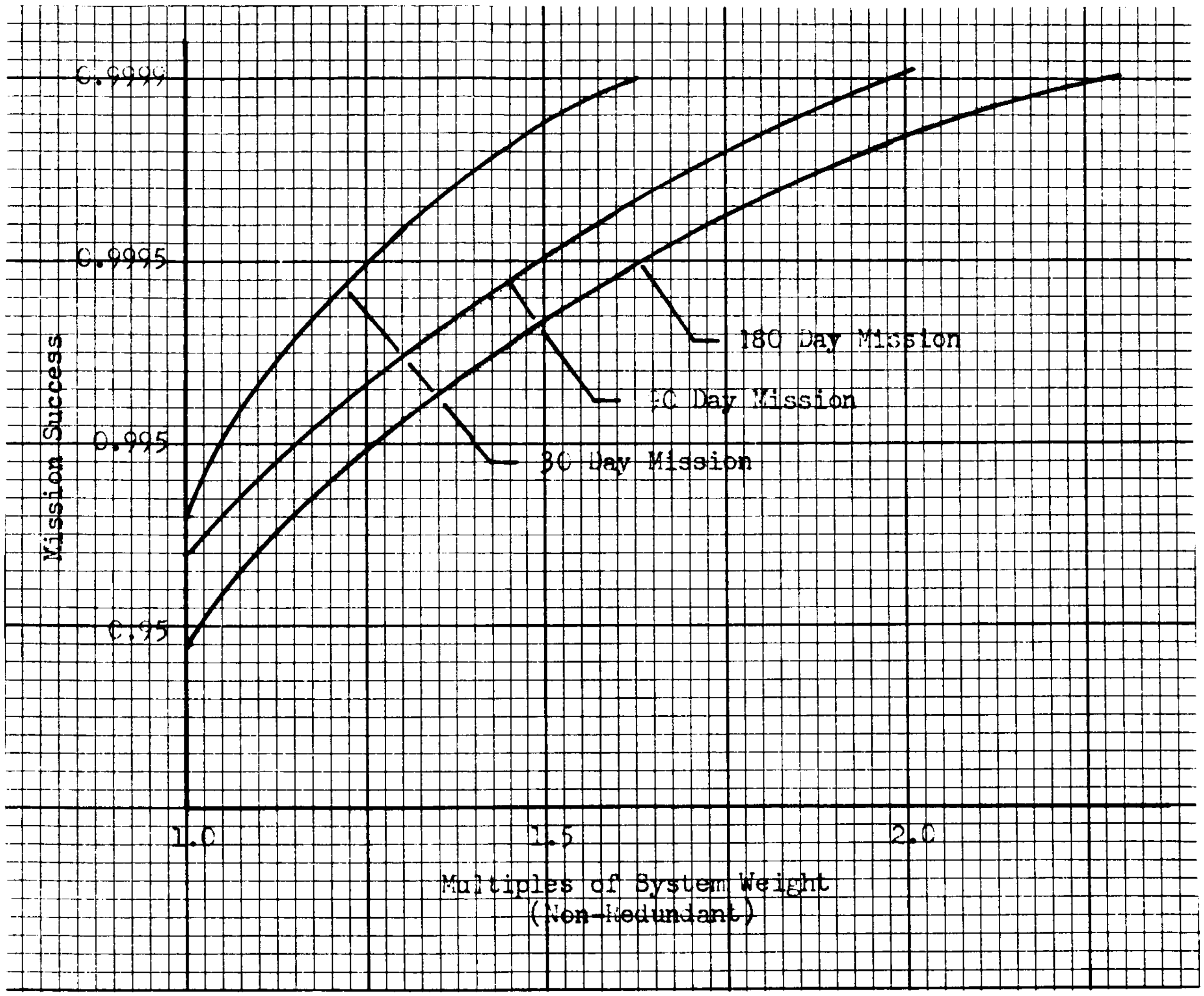


Figure 78. Weight Penalties Imposed by Varying Reliability Goals and Mission Durations

3.0 FORECAST FOR NEXT QUARTER

The major effort during the next quarter will be directed toward the synthesis and analysis of integrated instrument package and vehicle thermal systems. The parametric study and trade-off analysis at the component and subsystem level will be continued but at lower level of effort. A minimal effort will be conducted to complete the survey of developmental trends in the review of astrionic equipment and temperature regulation concepts, and in defining the astrionic equipment and equipment functions for the selected future missions.

Competitive integrated thermal control systems for thermally conditioning electronic packages for the various Saturn mission/vehicle combination will be established. For the competitive systems, sufficient data will be provided to permit rational evaluation and selection of optimum or near optimum systems. Among the significant system data to be provided includes: weight, reliability, power requirements, surface area and space requirements, developmental requirements and probability of successful development within the required time period.

System analysis to be conducted include design point and off-design performance analysis, reliability analysis and weight analysis. The performance analysis will consist of material and heat balances to establish efficiencies, flow rates, pressure drop, process material input-output rates, operating temperatures, cooling and/or heat requirements and power requirements. Reliability analysis will be made of the various approaches to achieving the desired reliability goals which are consistent with the overall mission success. The weight analysis will be made to provide total system weight as well as weight breakdown which will include: basic hardware weight, expendable weight, power penalty weight, and other possible penalty weights. These analyses will provide the data for system trade-off or comparative analysis.

At the component and subsystem level, the major effort will be devoted to the temperature regulation or control concepts and to establishing the regions of applicability of heat sinks. In conjunction with this effort, the heat pipe theory and concepts will be investigated since heat pipes appear to be highly feasible for certain applications.

4.0 REFERENCES

1. Quarterly Report No. 1, A Study of Electronic Packages Environmental Control Systems and Vehicle Thermal Systems Integration (Contract NAS 8-20320) NAA S&ID, SID 66-1277 (29 July 1966).

2. McCue, G. A., Program For Determining The Thermal Environment and Temperature History of Orbiting Space Vehicles, Aeronautical Systems Division, ASD-TN 61-83 (7 March 1962).

3. Nussberger, A. A., Power Systems Evaluation of Solar Versus Isotope for Early Manned Mars and/or Venus Flyby Missions, NAA S&ID, SID 65-1115 (August 1965).

4. Root, M. W., Application of Saturn/S-IVB/Apollo Systems to Planetary Exploration, presented to: Symposium on Post Apollo Space Exploration, American Astronautical Society (May 1965) Douglas Missile & Space Systems Division, Paper No. 3645

5. Dowdy, W. L. and W. H. Moritz. Utilization of Spent Launch Vehicle Stages in Support of Earth Orbital Missions, Final Report. NAA S&ID, SID 65-468 (31 May 1965).

6. Reithmaier, L. and B. Wendrow. Environmental Control and Life Support Systems Final Report, Preliminary Definition Phase Apollo Extension System. NAA S&ID, SID 65-1523 (17 December 1965).

7. Schneider, E. et al. Transient Analysis of Apollo Coldplate Network, NAA S&ID, SID 66-1383 (12 August 1966)

8. Plumlee, H. R., and D. A. Peterman, Infrared Microradiometer Thermal Studies of Operating Transistors, Texas Instruments, Inc.

9. Thermal Test Report, Cooling Power Supply Electronics With Fluorocarbon Dielectric Fluids, NAA-Autonetics, Autonetics TM-65-346-03-133 (3 June 1966).

10. The Heat Pipe, A Unique and Versatile Device For Heat Transfer Applications, RCA, Electronic Component and Devices Div., RCA Ref. 994-619.

11. Bacha, C.P., F. I. Honea, and D. Watanabe. Temperature Control Systems For Space Vehicles. Aeronautical Systems Division, ASD-TDR 62-493, Part I (6 April 1962).

12. Fixler, S. Z., "Satellite Thermal Control Using Phase-Change Materials," Journal of Spacecraft and Rockets, Vol. 3, No. 9 (Sept. 1966) pp. 1362-1368.

13. MacKay, D. B., Space Radiator Analysis and Design, Aeronautical Systems Division, ASD TR 61-30, Part II (17 February 1962).

14. Jacob, M., Heat Transfer, Vol. I, John Wiley and Sons, Inc., New York, 6th printing, 1958.

15. Eckert, E. R. G., and R. M. Drake, Jr. Heat and Mass Transfer, McGraw-Hill Book Company, Inc., New York, 1959.

16. Giedt, W. H., Principles of Engineering Heat Transfer, D. Van Nostrand Company, Inc., Princeton, N. J., August 1957.

17. McAdams, W. H., Heat Transmission, McGraw-Hill Book Company, Inc., New York, 3rd edition, 1954.

APPENDIX A

DESIGN OF GLYCOL COOLER USING HYDROGEN VENT GAS

Hydrogen Inlet Conditions

Temperature = 42°R
Pressure = 3.5 psia
Able to use with subcooling to 29.5°R
No entrained liquid in hydrogen vent gas
Triple point temperature = 24.8°R
Triple point temperature = 1.02 psia

Cooling Fluid Inlet and Outlet Conditions

Temperature in = 40°F (500°R)
Temperature out = 20°F (480°R)
Pressure drop in cooler = 0.5 psia
Pressure in lines at least 2 psi above vapor pressure
Composition 40% water + 60% ethylene glycol
Glycol flow = 475 lb/hr (2 KW heat rejection)

Design Criteria

To insure that under no conditions does the heat exchanger freeze up or congeal (assumed -40°F) the wall temperature is always maintained above this temperature. The specific method involved is to see what ratio of glycol to hydrogen hA's is required at the design point to obtain this wall temperature at selected points through the exchanger and to use three (3) times this value for design point calculations to give a safety factor. The glycol side must have no welds within the exchanger. The cooler will be welded into the glycol circuit.

Hydrogen Properties

T°R	Enthalpy	Viscosity	$Pr^{2/3}$	C_p
42	325 Btu/lb	.0036 lb hr ft	.8 approx.	2.55 Btu/(lb)(°F)
141.5	576	.0090	.8	2.55
234	828	.0124	.8	2.90
317.5	1079	.0153	.8	3.14
395	1331	.0177	.8	3.30

Cooling Fluid Properties

40°F	ρ = 69 lb/cu ft	22	17.0	.72
20°F	ρ = 69 lb/cu ft	38	23.5	.70
0°F	ρ = 69 lb/cu ft	68	35.0	.68

The wide variation in hydrogen properties makes necessary the separation of the exchanger into different areas to be calculated separately.

Proposed Design

A single uniform glycol tube of aluminum .030 wall thickness .875 O.D. with .030 fins; initially the fins to be 1/4" to 1/2" in length which is equivalent to 8 fins around the interior and brazed in place.

$$\text{Heat transfer area} = 2 \times 8 \times .28 + .81 = 4.48 + 2.55 = 7.03\ \square''/''\ \text{length}$$
$$= .586\ \square'/'\ \text{length}$$

$$\text{Thru flow area} = .81^2 \times \pi/4 - 8 \times .28 \times .03 = .515 - .067 = .498\ \square''$$

$$\text{Glycol } G = \frac{475 \times 144}{.498} = 137{,}200\ \text{lb/hr}/\square' = 38\ \text{lb/sec}/\square'$$

$$Re = \frac{137{,}200 \times .25}{38 \times 12} = 75.2 \quad \text{assumes film } T = 20°F$$

$$@\ Re = 75.2 \quad St\,Pr^{2/3} = .020 \quad f = .22$$

$$h_G = G\,C_p\,(St\,Pr^{2/3})\,Pr^{-2/3} = 137{,}200 \times .70 \times .020 \times 1/23.5 = 82$$

$$\Delta P/ft = \frac{G^2 \times f \times A}{9275 \times \quad \times A_c} = \frac{38 \times 38 \times .22 \times 7.03 \times 12}{9275 \times 69 \times .498} = .084\ \text{psi/ft}$$

$$\text{Heat Transfer Area Hydrogen} = .875 \times \pi = 2.74\ \square''/''$$

$$\text{Area ratio} \quad \frac{7.03}{2.74} = 2.57 \qquad \frac{.586}{2.57} = .228\ \square'/'$$

For the purposes of calculation the heat exchanger will be broken into four exchangers of equal cooling capacity and an initial hydrogen in temperature of 42°R and hydrogen exit temperature of 395°R assumed.

Permissible Hydrogen Coefficient

For the wall temperature to be -40°F (the assumed freezing limit) the ratio of hot side divided by cold side temperature differences are calculated. A parallel flow heat exchanger is assumed.

Initial $\quad \dfrac{500-420}{420-42} = \dfrac{80}{378} = \dfrac{1}{4.72}$

1st 1/4 $\quad \dfrac{495-420}{420-141.5} = \dfrac{75}{278.5} = \dfrac{1}{3.71}$

2nd 1/4 $\quad \dfrac{490-420}{420-234} = \dfrac{70}{186} = \dfrac{1}{2.66}$

3rd 1/4 $\quad \dfrac{485-420}{420-317.5} = \dfrac{65}{102.5} = \dfrac{1}{1.58}$

Final $\quad \dfrac{480-420}{420-395} = \dfrac{60}{25} = \dfrac{1}{.415}$

The permissible hydrogen coefficient can be determined by taking the glycol coefficient and correcting this for the area ratio (2.57) the temperature ratio (1/4.72) and the reciprocal of the safety factor (1/3).

Initial $h_h = \frac{82 \times 2.57}{3 \times 4.72} = 14.9$ 3rd 1/4 $h_h = \frac{82 \times 2.57}{3 \times 1.58} = 44.4$

1st 1/4 $h_h = \frac{82 \times 2.57}{3 \times 3.71} = 18.9$ Final $h_h = \frac{82 \times 2.57}{3 \times .415} = 169$

2nd 1/4 $h_h = \frac{82 \times 2.57}{3 \times 2.66} = 26.3$

Assuming these coefficients can be obtained and using the average hydrogen coefficient, the UA and NTU's per linear foot of exchanger of the first quarter of the overall exchanger are calculated. The effectiveness required is calculated and the NTU's necessary to obtain this effectiveness is calculated by dividing the NTU's required by the NTU's per linear foot the length required for the first 1/4 of the exchanger is determined. This is repeated for the remaining quarters of the heat exchanger.

$(\frac{I}{UA})$ 1st 1/4/linear ft $= \frac{1}{82 \times .586} + \frac{1}{16.9 \times .228} = .0208 + .259 = \frac{1}{3.57}$

NTU's $= \frac{UA}{C_{min}} = \frac{3.57}{6.8 \times 2.55} = .216/\text{ft}$ req. $\epsilon = \frac{141.5 - 42}{500 - 42} = \frac{99.5}{458} = .217$

NTU's req. = .25 Length req. $= \frac{.25}{.216} = 1.15$ ft = 13.8 in

$(\frac{I}{UA})$ 2nd 1/4/linear ft $= .0208 + \frac{2}{(18.9 + 26.3) \times .228} = .0208 + .193 = \frac{1}{4.67}$

NTU's $= \frac{UA}{C_{min}} = \frac{4.67}{6.8 \times 2.72} = .252/\text{ft}$ req. $\epsilon = \frac{234 - 141.5}{495 - 141.5} = \frac{92}{353} = .262$

NTU's req. = .305 Length req. $= \frac{.305}{.252} = 1.21$ ft = 14.5 in

$(\frac{I}{UA})$ 3rd 1/4/linear ft $= .0208 + \frac{2}{(26.3 + 44.4) \times .228} = .0208 + .124 = \frac{1}{6.9}$

NTU's $= \frac{UA}{C_{min}} = \frac{6.9}{6.8 \times 3.02} = .336/\text{ft}$ req. $\epsilon = \frac{317.5 - 234}{490 - 234} = \frac{83.5}{256} = .326$

NTU's req. = .392 Length req. $= \frac{.392}{.336} = 1.17$ ft = 14.0 in

$(\frac{1}{UA})$ 4th 1/4/linear ft $= .208 + \dfrac{2}{(44.4 + 169) \times .228} = .0208 + .0938 = \dfrac{1}{8.62}$

NTU's $= \dfrac{UA}{C_{min}} = \dfrac{8.62}{6.8 \times 3.22} = .393/\text{ft}$ req. $\epsilon = \dfrac{395 - 317.5}{485 - 317.5} = \dfrac{77.5}{167.5} = .462$

NTU's req. = .625 Length req. $= \dfrac{.625}{.393} = 1.59$ ft = 19.2"

Overall Length = 13.8 + 14.5 + 14.0 + 19.2 = 61.5" use 68"

Glycol Pressure Drop = .084 x 68 x 1/12 = .475 lb/cu in

Glycol Film Temperatures

Initial 500 -42 = 458° $\dfrac{458}{2(1 + 3 \times 4.72)} = 15.1°F$

40°F -15.1 = 24.9°F film temperature

1st 1/4 495 - 141.5 = 353.5°F $\dfrac{353.5}{2(1 + 3 \times 3.72)} = 14.5°F$

35°F -14.5°F = 20.5°F film temperature

2nd 1/4 490 -234 = 256 $\dfrac{256}{2(1 + 3 \times 2.66)} = 14.6°F$

30°F -14.6 = 15.4°F film temperature

3rd 1/4 485 - 317.5 = 167.5 $\dfrac{167.5}{2(1 + 3 \times 1.58)} = 14.5°F$

25°F -14.5 = 10.5°F film temperature

Final 480 -395 = 85 $\dfrac{85}{2(1 + 3 \times .416)} = 18.9°F$

20°F -18.9 = 1.1°F film temperature

Initial assumed film temperature = 20°F would give lower glycol coefficients and thus require more area in last portion of heat exchanger. This is sufficiently close so that calculation can proceed with hydrogen side of heat exchanger.

Hydrogen Film Temperatures

Initial Wall Temp.	500 -2 (15.1) = 469.8°R (9.8°F)
1/4	495 -2 (14.5) = 466°R (6.0°F)
1/2	490 -2 (14.6) = 460.8°R (.8°F)
3/4	485 -2 (14.5) = 456°R (-4°F)
Final	480 -2 (18.9) = 442.2°R (17.8°F)

Hydrogen Film Temperatures and Physical Properties

		μ	$PR^{2/3}$	C_p
Initial	(469.8 + 42).5 = 255.9°R	.0132	.8	2.94
1/4	(466 + 141.5).5 = 303.7°R	.0150	.8	3.10
1/2	(460.8 + 234).5 = 347.4°R	.0163	.8	3.20
3/4	(456 + 317.5).5 = 386.7°R	.0175	.8	3.28
Final	(442.2 + 395).5 = 418.6°R	.0185	.8	3.33

$(St\ Pr)^{2/3}$ G Calculation Use $h = G\ C_p\ (St\ Pr^{2/3})Pr^{-2/3}$

Initial	$14.9 = G\ 2.94\ (St\ Pr^{2/3})\ 1.25$	$(St\ Pr^{2/3})\ G = 4.05$
1/4	$18.9 = G\ 3.10\ (St\ Pr^{2/3})\ 1.25$	$(St\ Pr^{2/3})\ G = 4.88$
1/2	$26.3 = G\ 3.20\ (St\ Pr^{2/3})\ 1.25$	$(St\ Pr^{2/3})\ G = 6.57$
3/4	$44.4 = G\ 3.28\ (St\ Pr^{2/3})\ 1.25$	$(St\ Pr^{2/3})\ G = 10.8$
Final	$169 = G\ 3.33\ (St\ Pr^{2/3})\ 1.25$	$(St\ Pr^{2/3})\ G = 40.5$

Calculate Required Dimensions of Hydrogen Passage

The hydrogen heat transfer coefficient was used to calculate the required $(St\ Pr^{2/3})G$ value. An I.D. for the hydrogen is selected by trial and error such that the resulting G value gives a Reynolds number and consequently $(St\ Pr^{2/3})$ value such that the required $(St\ Pr^{2/3})$ G value is obtained. This is repeated for each portion of the heat exchanger.

$$\text{Hydrogen Weight Flow} = \frac{2 \times 3413}{1331 - 325} = 6.8 \text{ lb/hr}$$

Initial I.D. = 1.312 O.D. = .875 .437 = Hyd Dia.

$$A = \left[(1.312)^2 - (.875)^2\right]\frac{\pi}{4} = 1.352 - .601 = .751$$

$$G = \frac{6.8 \times 144}{.751} = 1305 \qquad Re = \frac{.437 \times 1305}{12 \times .0132} = 3610 \qquad f = .0091$$

$$(St\ Pr^{2/3}) = .0031$$

$(St\ Pr^{2/3})\ G = .0031 \times 1305 + 4.05$ check

1/4 I.D. = 1.250 O.D. .875 ; .375 = Hyd Dia.

$$A = \left[(1.25)^2 - (.875)^2\right]\frac{\pi}{4} = 1.228 - .601 = .627$$

$$G = \frac{6.8 \times 144}{.627} = 1560 \qquad Re = \frac{.375 \times .1560}{12 \times .0150} = 3250$$

f = .0095

$(St\ Pr^{2/3}) = .0031$

$(St\ Pr^{2/3})\ G = .0031 \times 1560 = 4.84$ close

1/2 I.D. = 1.155 O D. .875 ; .280 = Hyd Dia.

$$A = \left[(1.155)^2 - (.875)^2\right]\frac{\pi}{4} = 1.046 - .601 = .445$$

$$G = \frac{6.8 \times 144}{.445} = 2200 \qquad Re = \frac{.28 \times 2200}{12 \times .0163} = 3150$$

f = .0094

$(St\ Pr^{2/3}) = .003$

$(St\ Pr^{2/3})\ G = .003 \times 2200 = 6.60$ close

3/4 I.D. = 1.050 O.D. = .875 ; .175 = Hyd Dia.

$$A = \left[(1.05)^2 - (.875)^2\right]\frac{\pi}{4} = .865 - .601 = .264$$

$$G = \frac{6.8 \times 144}{.264} = 3710 \qquad Re = \frac{.175 \times 3710}{12 \times .0175} = 3090$$

f = .0093

$(St\ Pr^{2/3}) = .0029$

$(St\ Pr^{2/3})\ G = .0029 \times 3710 = 10.76$ close

A smoothed plot of required $(St\ Pr^{2/3})$ G stops at .93 through the exchanger at a value of .23 calculated from .75 to .93.

.93 I.D. = .960 O.D. = .875 ; .085 = Hyd Dia.

$$A = \left[(.96)^2 - (.875)^2\right]\frac{\pi}{4} = .724 - .601 = .123$$

$$G = \frac{6.8 \times 144}{.123} = 7960 \qquad Re = \frac{.085 \times 7960}{12 \times .0181} = 3110$$

f = .0093

$(St\ Pr^{2/3}) = .0029$

$(St\ Pr^{2/3})\ G = .0029 \times 7960 = 23$ check

For the last 7%, the hydrogen side stays constant in cross section and heat transfer coefficient, i.e. 96 Btu/(hr)(sq ft)(°F). This will require an increase in area of the last 7%. This can be determined as follows:

$$U \ @ \ .93 = \frac{1}{96} + \frac{1}{82x2.57} = .0104 + .00475 = .01515$$

$$\text{Previous U final} = \frac{1}{167 \times 1} + \frac{1}{82 \times 2.57} = .006 + .00475 = .01075$$

.01515/.01075 = 1.41 average increase = (1.41 + 1.0) x .5 = 1.21

19.2 x .07 + .25 = 5.4   5.4 x 1.21 = 6.5   6.5-5.4 = 1.1"

increase overall length 1" i.e. 68" goes to 69"

The hydrogen pressure drop is calculated for both a 10 psi initial pressure and a 35 psi initial pressure. The calculations show that at or below 3.5 psi the exchanger will have a reduced hydrogen flow and capacity.

1st 1/4   Initial Pressure 10 psia   1#/□" = 27.7" H_2O

42 + 141.5 = 183.5; 183.5 / 2 = 91.7°R - Use 100°R

$$\rho = .0765 \times \frac{10}{14.7} \times \frac{2}{29} \times \frac{520}{100} = .0186$$

$$\left(\frac{1432}{3600}\right)^2 = .158$$

G=1305 + 1560 = 2865; 2865 / 2 = 1432

.751 + .627 = 1.378; 1.378 / 2 = .689

$$\Delta P = \frac{.158}{25.6}\left(\frac{.0765}{.0186}\right)\left(.6 + .0091\,\frac{13.8 \times 2.77 \times \left[1 + \frac{1.32}{.885}\right]}{.689}\right)$$

277.000 − .048 = 276.952

$$\Delta P = (.0254)(.6+1.27) = .048" H_2O$$

2nd 1/4

141.5 + 234 = 375.5; 375.5 / 2 = 187.7   Use 190°R

$$= .0765 \times \frac{10}{14.7} \times \frac{2}{29} \times \frac{520}{190} = .0089$$

1560 + 2200 = 3760; 3760 / 2 = 1880

.627 + .445 = 1.072; 1.072 / 2 = .536

$$\left(\frac{1880}{3600}\right)^2 = .273$$

1.25 + 1.155 = 2.405; 2.405 / 2 = 1.2

0091 + 0094 = 0185; 0185 / 2 = .0093

$$\Delta P = \frac{.273}{25.6}\left(\frac{.0765}{.0089}\right)\left(.0093\,\frac{14.5 \times 2.77 \times}{.536}\left[1 + \frac{1.2}{.8575}\right]\right)$$

276.952 − .146 = 276.806

$$\Delta P = (.0919)(1.65) = .146" H_2O$$

3rd 1/4

$$\frac{234 + 317.5}{2} = \frac{551.5}{2} = 276$$

$$\rho = .0765 \times \frac{10}{14.7} \times \frac{2}{29} \times \frac{520}{276} = .0067$$

$$\frac{2200 + 3710}{2} = \frac{5910}{2} = 2955 \qquad \frac{.445 + .264}{2} = \frac{.709}{2} = .355$$

$$\left(\frac{2955}{3600}\right)^2 = .675$$

$$\Delta P = \frac{.675}{25.6}\left(\frac{.0765}{.0067}\right)\left(.0095\ \frac{14.0 \times 2.77 \times \left[1 + \frac{1.1}{.875}\right]}{.355}\right)$$

$$\frac{1.155 + 1.050}{2} = \frac{2.205}{2} = 1.1$$

$$\Delta P = (.301)(2.3) = .70''H_2O$$

$$276.806 - .70 = 276.106$$

The 4th quarter of the exchanger will be broken into two parts the first 18% the second 7%.

4th 18%

$$\begin{array}{r} 395 \\ 317.5 \\ \hline 81.5 \end{array} \times \frac{18}{25} = 58.8$$

$$\rho = .0756\ \frac{9.8\ \text{est.}}{14.7} \times \frac{2}{29} \times \frac{520}{346.9} = .0053$$

$$\frac{3710 + 7960}{2} = \frac{11670}{2} = 5835 \qquad \frac{.264 + .123}{2} = \frac{.387}{2} = .194$$

$$\frac{58.8}{2} + 3.17.5 = 346.9 \qquad \left(\frac{5835}{3600}\right)^2 = 2.63$$

$$\frac{.96 + 1.05}{2} = \frac{2.01}{2} = 1.005$$

$$\Delta P = \frac{2.63}{25.6}\left(\frac{.0765}{.0053}\right)\left(.0093\ \frac{13.8 \times 2.77 \left[1 + \frac{1.005}{.875}\right]}{.194}\right)$$

$$\Delta P = (1.48)(3.94) = 5.82 \qquad \frac{273}{277} \times 10 = 9.85 \text{ very close}$$

$$276.11 - 5.82 = 270.29$$

4th 7%

$$\begin{array}{r} 317.5 \\ 58.8 \\ \hline 365.3 \\ 395.0 \\ \hline 2\ \overline{)761.3} \\ 380.7 \end{array}$$

$$\rho = .0765\ \frac{9.6\ \text{est.}}{14.7} \times \frac{2}{29} \times \frac{520}{380.7} = .0047$$

$$\left(\frac{7960}{3600}\right)^2 = 4.90$$

$$\Delta P = \frac{4.9}{25.6}\left(\frac{.0765}{.0047}\right)\left(.0093\ \frac{6.5 \times 2.77 \left[1 + \frac{.96}{.875}\right]}{.123}\right)$$

$$270.29 - 8.90 = 261.39\ ''H_2O \text{ final pressure}$$

$$\Delta P = (3.11)(2.86) = 8.90$$

Pressure Drop Hydrogen
Initial Pressure 3.5 psia Triple Point 1.02 psia 3.5 psi = 96.8"H_2O

1st 1/4 $\rho = .0765 \times \frac{3.5}{14.7} \times \frac{2}{29} \times \frac{520}{100} = .00652$

$$\Delta P = \frac{.158}{25.6}\left(\frac{.0765}{.00652}\right)\left(.6 + .0091\ \frac{13.8 \times 2.77 \times \left[1 + \frac{1.2}{.875}\right]}{.689}\right)$$

$$\Delta P = (.073)(1.87) = .137"H_2O$$

2nd 1/4 $\rho = .0765 \times \frac{3.5}{14.7} \times \frac{2}{29} \times \frac{520}{190} = .00311$

$$\Delta P = \frac{.273}{25.6}\left(\frac{.0765}{.00311}\right)\left(.0093\ \frac{14.5 \times 2.77 \times \left[1 + \frac{1.2}{.875}\right]}{.536}\right)$$

$$\Delta P = (.263)(1.65) = .433"H_2O$$

3rd 1/4 $\rho = .0765 \times \frac{3.45}{14.7}\text{ est.} \times \frac{2}{29} \times \frac{520}{276} = .0017$

$$\Delta P = \frac{.675}{25.6}\left(\frac{.0765}{.0017}\right)\left(.0095\ \frac{14.0 \times 2.77 \times \left[1 + \frac{1.1}{.875}\right]}{.355}\right)$$

$$\Delta P = (.875)(2.30) = 2.01"H_2O$$

$$\begin{array}{r} 96.800 \\ \underline{.137} \\ 96.663 \\ \underline{.433} \\ 96.230 \\ \underline{2.01} \\ 94.220 \\ \underline{19.40} \\ 74.82 \\ \underline{44.5} \\ 30.32 \end{array}$$

The 4th quarter of the exchanger will be broken into two parts. The first 18% the second 7%.

4th 18% $\rho = .0765\ \frac{3.1\text{ est.}}{14.7} \times \frac{2}{29} \times \frac{520}{347} = .00161$

$$\Delta P = \frac{2.63}{25.6}\left(\frac{.0765}{.0016}\right)\left(.0093\ \frac{13.8 \times 2.77 \times \left[1 + \frac{1.0}{.875}\right]}{.194}\right)$$

$$\Delta P = (4.91)(3.94) = 19.4"H_2O$$

$$\begin{array}{r} 94.2 \\ \underline{9.7} \\ \frac{84.5}{96.8} \times 3.5 = 3.05 \end{array}$$

close

4th 7% $\rho = .0765\ \frac{2.0\text{ est.}}{14.7}\ \frac{2}{29} \times \frac{520}{380.7} = .00095$

$$\Delta P = \frac{4.9}{25.6}\left(\frac{.0765}{.00095}\right)\left(.0093\ \frac{6.5 \times 2.77\left[1 + \frac{.96}{.875}\right]}{.123}\right)$$

$$\Delta P = (15.5)(2.86) = 44.5$$

$$\begin{array}{r} 75 \\ \underline{22.2} \\ \frac{52.8}{96.8} \times 3.5 = 1.91 \end{array}$$

not too far off

Final pressure 30.3 "H_2O